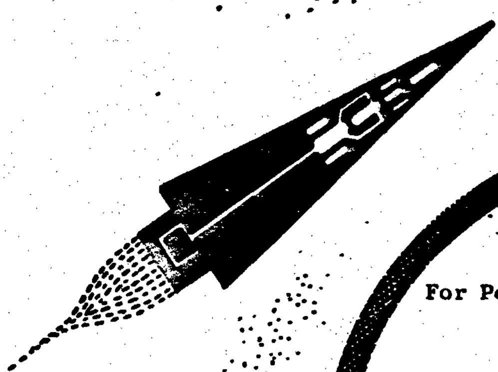
SPACE POWER AND PROPULSION SECTION



QUARTERLY REPORT #3

For Period Ending January 15, 1963

DYNAMIC SHAFT SEALS IN SPACE
Under Contract #AF 33(657)-8459

For

Aeronautical Systems Division

Wright Patterson Air Force Base, Ohio

SPACECRAFT DEPARTMENT
MISSILE and SPACE DIVISION
GENERAL ELECTRIC
CINCINNATI, OHIO

SPACE POWER AND PROPULSION SECTION

QUARTERLY PROJECT STATUS REPORT

JANUARY 15, 1962

DYNAMIC SHAFT SEALS IN SPACE

Under Contract #AF 33(657)-8459

RE-ENTRY SYSTEMS DEPARTMENT
MISSILE AND SPACE DIVISION
GENERAL ELECTRIC COMPANY
CINCINNATI 15, OHIO

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I. SUMMARY

The Space Power and Propulsion Section of the General Electric Company has been under contract to the Aeronautical Systems Division, Wright Patterson Air Force Base, Ohio, since April 15, 1962 for the development of dynamic shaft seals for space applications. The objective of this program is to acquire the techniques for sealing high speed rotating shafts under the operating conditions of high temperature liquid metals and vapors, the near-vacuum environments of space, and to provide long seal life.

A. The contract specifies the following requirements:

1. The fluid to be sealed shall be potassium.
2. The seals shall be operative at fluid temperatures from the melting point of the fluid selected to 1400°F.
3. The pressure on the fluid side of the seal shall be 15 psi and the external pressure shall be 10^{-6} mm Hg.
4. The speed of the rotating shaft shall be a maximum of 36,000 rpm.
5. The seal, or seal combinations, shall be designed for 10,000 hours of maintenance-free life.
6. The working fluid, potassium, shall be used as the seal lubricant.
7. The seal, or seal combinations, shall be capable of maintaining zero leakage - in the technical sense - under all conditions of operation.
8. The seals shall be designed for a 1.0 inch diameter shaft.

9. The seals shall be capable of operating in a zero "g" environment.

B. The seal evaluation shall consist of:

1. Preliminary experiments with water.
2. 100-hour operational screening test with liquid metal.
3. Thermal-cycling test with liquid metal.
4. 3000-hour life test with liquid metal.

This report covers progress during the quarter ending January 15, 1962. The main events of this reporting period are:

1. Manufacture of the rotating disk-squeeze seal configuration for water testing to 20,000 rpm was completed.
2. Testing of the rotating disk-squeeze seal configuration in water was initiated and its feasibility was proven.
3. Manufacture of the rotating disk seal configuration for water testing to 20,000 rpm was completed.
4. Testing of the rotating disk seal configuration in water was initiated and its feasibility proven.
5. Testing of the screw seal configuration in water and silicone fluid up to speeds of 35,000 rpm continued.
6. Evaluation of test results from the screw seal testing comparison of the results to analytical expectations was initiated.
7. Evaluation of preliminary results from both the rotating disk seal and the rotating disk-squeeze seal configurations was initiated.

8. Design and placement of manufacturing order for the double disk seal configuration for 20,000 rpm water tests.

9. Design of the liquid metal seal test facility was completed.

10. Manufacturing orders of all major components in the liquid metal seal test facility were placed.

11. The design of the liquid metal seal test rig for operation at speeds up to 36,000 rpm was finalized.

12. The drive spindle end of the 36,000 rpm test rig has been sent out for manufacturing quotations to eight qualified vendors.

II. FLUID DYNAMIC TESTING

The water test rig for the rotating disk-squeeze seal configuration was manufactured this quarter. The configuration is shown in Figure 3. Set-up of the seal configuration on the 20,000 rpm test spindle was satisfactory. This spindle was previously used for the rotating housing-stationary disk seal configuration in Figure 2. Preliminary check-out was successful and testing is currently in progress. The test rig modifications which may be made to completely investigate this seal configuration and the test procedure to be followed is covered below.

A. Test Procedure for Squeeze-Seal

1. Build-up water seal test rig with the configurations specified in Table I..
2. Run the seal at the maximum possible sealing pressure for the following speeds and plot.

   5,000 ± 100 rpm

   10,000 ± 200 rpm

   15,000 ± 500 rpm

   0

   20,000 ± 1000 rpm

3. Measure the following parameters on Sanborn at each speed point.
   a. Speed
   b. Seal Pressure Profile
   c. Sealing Pressure

d. Seal Rim Pressure

e. Seal Torque

f. Seal Water Temperature, In

g. Seal Water Temperature, Out of Disk

h. Seal Water Temperature, Out.

4. Seal cooling water flow will be set at 2 gpm for all test configurations. In addition, test configurations shall be repeated with water flow set at 1 gpm.

The squeeze seal test configurations include variation of several significant parameters. These parameters are: rotating disk axial spacing changes on both the low pressure and high pressure side, changes of the minimum water level on the low pressure side of the disk, changes in the recirculating ring axial length, changes in squeeze annulus clearance and changes in the annulus bushing axial length. These configurations are identified by the following numbering sequence and Table II.

First digit	3	Represents squeeze seal.
Next two digits	00	Represents disk configuration, Part a.
Next digit	0	Represents r_L configuration, Part b.
Next digit	0	Represents recirculating ring configuration, Part c.
Next digit	0	Represents spacer-sleeve configuration, Part d.
Next digit	0	Represents bushing configuration, Part e.

Sample Configuration Number: 3-00-0000

TABLE I

TEST CONFIGURATION

SQUEEZE SEAL

Part a

Disk Config. No.	$\underline{S_L}$ S_L = Spacer + 0.031	$\underline{S_H}$ S_H Spacer + .097 - S_L	Spindle Configuration: Low Pressure Side Spacer P/N4012000-884	Spindle Configuration: High Pressure Side Spacer P/N4012000-886
1	.031	.066	No Spacer	No Spacer
2	.031	.166	No Spacer	(1)#1 Spacer (.100)
3	.031	.266	No Spacer	(2)#1 Spacer
4	.031	.366	No Spacer	(1)#2 Spacer (.300)
5	.031	.466	No Spacer	(1)#1 & (1)#2 Spacer
6	.031	.566	No Spacer	(2)#1 & (1)#2 Spacer
7	.031	.666	No Spacer	(2)#2 Spacer
8	.031	.766	No Spacer	(1)#1 & (2)#2 Spacer
9	.081	.016	#1 Spacer (.050)	No Spacer
10	.081	.116	#1 Spacer	(1)#1 Spacer
11	.081	.216	#1 Spacer	(2)#1 Spacer
12	.081	.316	#1 Spacer	(1)#2 Spacer
13	.081	.416	#1 Spacer	(1)#1 & (1)#2 Spacer
14	.081	.516	#1 Spacer	(2)#1 & (1)#2 Spacer
15	.081	.616	#1 Spacer	(2) #2 Spacer
16	.081	.716	#1 Spacer	(1)#1 & (2)#2 Spacer
17	.131	.066	#2 Spacer (.100)	(1)#1 Spacer
18	.131	.166	#2 Spacer	(2)#1 Spacer
19	.131	.266	#2 Spacer	(1)#2 Spacer
20	.131	.366	#2 Spacer	(1)#1 & (1)#2 Spacer
21	.131	.466	#2 Spacer	(2)#1 & (1)#2 Spacer
22	.131	.566	#2 Spacer	(2)#2 Spacer

TABLE I (Continued)

Part a

Disk Config. No.	S_L S_L = Spacer + 0.031	S_H S_H Spacer + .097 - S_L	Spindle Configuration Low Pressure Side Spacer P/N4012000-884	High Pressure Side Spacer P/N4012000-886
23	.131	.666	#2 Spacer	(1)#1 & (2)#2 Spacer
24	.231	.066	#1 & #3 Spacer	(2)#1 Spacer
25	.231	.166	#1 & #3 Spacer	(1)#2 Spacer
26	.231	.266	#1 & #3 Spacer	(1)#1 & (1)#2 Spacer
27	.231	.366	#1 & #3 Spacer	(2)#1 & (1)#2 Spacer
28	.231	.466	#1 & #3 Spacer	(2)#2 Spacer
29	.231	.566	#1 & #3 Spacer	(1)#1 & (2)#2 Spacer
30	.231	.666	#1 & #3 Spacer	(2)#1 & (2)#2 Spacer
31	.331	.066	#4 Spacer (.300)	(1)#2 Spacer
32	.331	.166	#4 Spacer	(1)#1 & (1)#2 Spacer
33	.331	.266	#4 Spacer	(2)#1 & (1)#2 Spacer
34	.331	.366	#4 Spacer	(2)#2 Spacer
35	.331	.466	#4 Spacer	(1)#1 & (2)#2 Spacer
36	.331	.566	#4 Spacer	(2)#1 & (2)#2 Spacer
37	.431	.006	(1)#2 & (1)#4 Spacer	(1)#1 & (1)#2 Spacer
38	.431	.166	(1)#2 & (1)#4 Spacer	(2)#1 & (1)#2 Spacer
39	.431	.266	(1)#2 & (1)#4 Spacer	(2)#2 Spacer
40	.431	.366	(1)#2 & (1)#4 Spacer	(1)#1 & (2)#2 Spacer
41	.431	.466	(1)#2 & (1)#4 Spacer	(2)#1 & (2)#2 Spacer
42	.631	.066	(1)#1,(1)#2,(1)#3 & (1)#4	(2)#2 Spacer
43	.631	.166	(1)#1,(1)#2,(1)#3 & (1)#4	(1)#1 & (2)#2 Spacer
44	.631	.266	(1)#1,(1)#2,(1)#3 & (1)#4	(2)#1 & (2)#2 Spacer

TABLE I

Part b

Housing Configuration No.	r_L
1	1.22
2	0.84

Part c

Recirculating Ring Configuration No.	Recirculating Ring P/N4012000-878
1	#1 Ring (1.006)
2	#2 Ring (.682)
3	#3 Ring (.374)

Part d

Spacer-Sleeve Configuration No.	Spacer-Sleeve P/N4012000-881
1	#1 Spacer (2.496)
2	#2 Spacer (2.446)
3	#3 Spacer (2.393)

Part e

Bushing Configuration No.	Bushing P/N4012000-877
1	#1 Bushing (.905)
2	#2 Bushing (1.219)
3	#3 Bushing (1.527)

B. <u>Test Procedure For Rotating Disk Seal</u>

1. Build up water seal test rig with the configurations specified in Table I.

2. Run the seal at the maximum possible sealing pressure for the following speeds and plot.

   $5{,}000 \pm 100$ rpm

   $10{,}000 \pm 200$ rpm

   $15{,}000 \pm 500$ rpm

   $20{,}000 \, {}^{+\,0}_{-\,1000}$ rpm

3. Measure the following parameters on Sanborn at each speed point.

   a. Speed
   b. Seal Pressure Profile
   c. Sealing Pressure
   d. Seal Rim Pressure
   e. Seal Torque
   f. Seal Water Temperature, In
   g. Seal Water Temperature, Out

4. Seal cooling water flow will be set at 2 gpm for all test configurations. In addition, test configurations 1, 11, 22 and 33 shall be repeated with water flow set at 1 gpm.

5. Repeat items 2, 3 and 4 with pressure tap #5 maintained at atmospheric pressure.

TABLE II

TEST CONFIGURATIONS

Disk Config. No.	Rotating Disk S_L S_L = Spacer + 0.031	S_H S_H = Spacer + 0.312 - S_L - Insert	Spindle Configuration: Low Pressure Side Spacer P/N4012000-884	Spindle Configuration: High Pressure Side Spacer P/N 4012000-886 Insert P/N 4012000-941
1	.031	.031	No Spacer	No Spacer, #2 insert
2	.031	.131	No Spacer	(1)#1 Spacer, #2 Insert
3	.031	.181	No Spacer	(1)#1 Spacer, #1 Insert
4	.031	.281	No Spacer	No Spacer or Insert
5	.031	.381	No Spacer	(1)#1 Spacer
6	.031	.581	No Spacer	(1)#2 Spacer
7	.031	.781	No Spacer	(2)#1 & (1)#2 Spacer
8	.031	.981	No Spacer	(1)#1 & (2)#2 Spacer
9	.081	.031	#1 Spacer (.050)	No Spacer, #1 insert
10	.081	.131	#1 Spacer	(1)#1 Spacer, #1 Insert
11	.081	.231	#1 Spacer	No Spacer or Insert
12	.081	.331	#1 Spacer	(1)#1 Spacer
13	.081	.531	#1 Spacer	(1)#2 Spacer
14	.081	.731	#1 Spacer	(2)#1 & (1)#2 Spacer
15	.131	.031	#2 Spacer (.100)	(1)#1 Spacer, #2 Insert
16	.131	.131	#2 Spacer	(2)#1 Spacer, #2 Insert
17	.131	.181	#2 Spacer (.100)	No Spacer or Insert
18	.131	.281	#2 Spacer	(1)#1 Spacer
19	.131	.381	#2 Spacer	(2)#1 Spacer
20	.131	.581	#2 Spacer	(1)#1 & (1)#2 Spacer
21	.131	.781	#2 Spacer	(2)#2 Spacer
22	.231	.031	(1)#1 & (1)#2 Spacer	(2)#1 Spacer, #2 Insert
23	.231	.131	(1)#1 & (1)#3 Spacer	(1)#2 Spacer, #2 Insert

TABLE II (Continued)

Disk Config. No.	S_L S_L = Spacer + 0.031	S_H S_H - Spacer + 0.312 - S_L - Insert	Spindle Configuration Low Pressure Side Spacer P/N4012000-884	 High Pressure Side Spacer P/N4012000-886 Insert P/N4012000-941
24	.231	.181	(1)#1 & (1)#3 Spacer	(1)#1 Spacer
25	.231	.281	(1)#1 & (1)#3 Spacer	(2)#1 Spacer
26	.231	.381	(1)#1 & (1)#3 Spacer	(1)#2 Spacer
27	.231	.581	(1)#1 & (1)#3 Spacer	(2)#1 & (1)#2 Spacer
28	.231	.781	(1)#1 & (1)#3 Spacer	(1)#1 & (2)#2 Spacer
29	.331	.031	#4 Spacer (.300)	(1)#2 Spacer, #2 Insert
30	.331	.081	#4 Spacer	(1)#1 Spacer
31	.331	.181	#4 Spacer	(2)#1 Spacer
32	.331	.281	#4 Spacer	(1)#2 Spacer
33	.331	.381	#4 Spacer	(1)#1 & (1)#2 Spacer
34	.331	.581	#4 Spacer	(2)#2 Spacer
35	.331	.781	#4 Spacer (.300)	(2)#1 & (2)#2 Spacer
36	.431	.031	(1)#2 & (1)#4 Spacer (.400)	(1)#1 & (1)#2 Spacer, #2 Insert
37	.431	.081	(1)#2 & (1)#4 Spacer	(2)#1 Spacer
38	.431	.181	(1)#2 & (1)#4 Spacer	(1)#2 Spacer
39	.431	.281	(1)#2 & (1)#4 Spacer	(1)#1 & (1)#2 Spacer
40	.431	.381	(1)#2 & (1)#4 Spacer	(2)#1 & (1)#2 Spacer
41	.431	.581	(1)#2 & (1)#4 Spacer	(1)#1 & (2)#2 Spacer
42	.581	.031	(1)#2,(1)#3 & (1)#4 Spacers	(1) #2 Spacer
43	.581	.131	(1)#2,(1)#3 & (1)#4 Spacer	(1)#1 & (1)#2 Spacer
44	.581	.231	(1)#2,(1)#3 & (1)#4 Spacer	(2)#1 & (1)#2 Spacer
45	.581	.331	(1)#2,(1)#3 & (1)#4 Spacer	(2)#2 Spacer
46	.581	.531	(1)#2,(1)#3 & (1)#4 Spacer	(2)#1 & (2)#2 Spacer

Testing of the rotating disk-squeeze seal and the rotating disk seal configurations with water as a working fluid has been performed for determination of the following:

a. What is the maximum sealing pressure possible for a given diameter seal?

b. What is the pressure profile within the seal?

c. What is the power requirement for a given diameter seal?

d. What is the required clearance between the rotating disk and stationary wall to minimize heating problems and maximize sealing capability?

e. What cooling flow is required to remove the heat generated by fluid friction?

The tests have definitely proved the feasibility of the rotating disk seal and the squeeze seal. Both of these seals demonstrated ability to seal liquid and gas. Preliminary evaluation of some of the test data has been made and is shown in Figures 17 through 20. Further evaluation of test data for these configurations is continuing.

C. Screw Seal

The screw seal test rig has been run to rotational speeds of 35,000 rpm. Operation was stable. The sealing coefficients obtained approximate those measured with a free floating sleeve. Since no rubbing occurred, measured friction factors were lower than with the free floating sleeve. The values obtained with a fixed sleeve more closely matched those of an unloaded journal bearing.

Figures 8 and 9 show the screw seal test set up and instrumentation set up. Figure 10 shows the screw configuration. A summary of screw seal test results is given in Figures 21 and 22.

Reference 7 describes the screw seal tests conducted with a free floating sleeve. This configuration proved to be unstable in the turbulent flow regime and it was not possible to reach the desired operating speed of 35,000 rpm. To correct this, the sleeve was mounted in ball bearings. The fixed sleeve configuration shown in Figure 3 is much more stable than the floating sleeve, and operation to 35,000 rpm was successful.

A summary of the test results for fixed sleeve operation is shown in Figure 4. The table includes the fluid being sealed, rotational speed, sealing pressure, length of wetted shaft and power consumed. In general, the power requirements were lower with the fixed sleeve than with the floating sleeve due to the fact that rubbing occurred with the floating sleeve.

1. Fixed Sleeve Sealing Coefficient

The measured sealing coefficients for fixed sleeve operation approximated those measured with the floating sleeve. The laminar sealing coefficient, however, approached a value of .34 for fixed sleeve operation rather than the value of .43 measured with the floating sleeve. This is not unexpected as the leakage flow term in the laminar sealing equation is a function of eccentricity. Since the free floating sleeve tended to center itself, i.e. zero eccentricity, the leakage back over the lands was a minimum. With the fixed sleeve, it is very difficult to exactly center the shaft. Any slight eccentricity will increase the land leakage flow and thus lower the sealing coefficient.

Figures 23 and 24 show the sealing coefficient as a function of Reynolds number. Data from both fixed and floating operation show correlation. The sealing equation has previously been shown to be of the form:

$$\frac{\Delta P \ \delta^2}{\mu \ VL} = K_1' + K_2' \ Re^n$$

The data of Figures 23 and 24 are approximated by the equation

$$\frac{\Delta P \ \delta^2}{\mu \ VL} \approx .34 \text{ for the laminar flow regime } (Re < 800).$$

The equation

$$\frac{\Delta P \ \delta^2}{\mu \ VL} \approx .34 + 1.8 \times 10^{-5} \ Re^{1.25}$$

for the turbulent regime ($Re > 3000$). Data is currently being taken in the transition region ($700 > Re > 3000$) using a 2 centistoke test fluid.

2. <u>Friction Factor Measurements</u>

Measured friction factors for fixed sleeve operation are shown in Figure 25. They are much lower than the friction factors found with the free floating sleeve reported in Reference 7. This is to be expected since rubbing occurred in free sleeve operation.

In the laminar operation, the measured friction factor is given by the relation

$$f = \frac{130}{Re^{3/2}}$$

and for Reynolds numbers above 6000 the data closely approximates the friction factor for an unloaded journal bearing. Below this, the values of friction factor appear to exhibit a slope of 3/2 as was measured in the laminar regime. This range of operation is currently being more thoroughly investigated with a 2 centistoke viscosity fluid.

3. Seal Breakdown

For each type of fluid tested there is a certain shaft speed for which the seal will start to leak slightly, perhaps a drop every two or three minutes. This phenomena occurs in both laminar and turbulent operation. Listed below are the operating conditions for the three types of test fluids at which leakage was first observed.

Fluid	Vapor Pressure in Hg	ΔP psi	N rpm	$\frac{\rho U^2}{2g}$ psi	Re
SF96-5	.028	36.1	12,700	13.4	592
SF96-0.65	.028	16	12,000	14.4	6300
Distilled Water	1.8	31	18,160	42.3	6356

Two reasons for this slight leakage are probable.

1. Axial backflow across screw lands.
2. Cavitation occurs.

Examination of the above table shows that slight leakage occurs in laminar flow at Reynolds number 592. While in turbulent flow for two different fluids, the Reynolds number at which leakage is observed is 6300. To determine if axial backflow is a function of Reynolds Number, the running clearance can be reduced and further tests performed to determine the lowest Reynolds Number that leakage occurs.

The other possible cause for the leakage is cavitation. When liquid enters a region where its static pressure is reduced to the vapor pressure, it is possible for it to boil. If this occurs, the vapor formed would most likely be expelled from the seal and condense in the atmosphere. The few drops of fluid observed leaking from the seal could then be the result of cavitation. If this were the case then, since the vapor pressures for all three test liquids are about the same, the leakage would occur at the same value of fluid velocity head, $\frac{\rho U^2}{2g}$. This is true for the silicone fluid but not the distilled water, so it would appear that cavitation was not the cause. However, the cavitation characteristics of different types of fluids vary tremendously, that is some fluids can sustain pressures much below their vapor pressure before cavitating while others cannot. The amount and type of impurities in the fluid can also have a considerable effect on their cavitation characteristics. Because of this it is not possible to entirely discount cavitation as a cause of seal breakdown.

III. MECHANICAL DESIGN

A. Double Disk Seal Configuration

Final design of the water test rig for the double disk seal configuration has been completed and is shown in Figure 7. The test configuration will be mounted on the 20,000 rpm test spindle. The manufacturing order for the configuration has been placed and scheduled delivery is February 25, 1963.

B. Liquid Metal Seal Test Spindle

Design of the 36,000 rpm high speed test rig for use with liquid metal has been progressing throughout this quarter. The seal drive spindle consists of anair turbine mounted on a ball bearing spindle similar to the present 20,000 rpm rig. The seal support spindle consists of a 3" diameter hollow shaft supported in three argon bearings. The spindle mounts the liquid metal seal configuration in an overhung position with vacuum connections at the end of the seal. This configuration is shown in Figure 13.

The design of the spindles for 36,000 rpm operation is very dependent upon the vibrational characteristics of the rotating shaft. Therefore, a complete vibration study was made of the selected shaft. Figure 14 summarizes the results of this study.

IV. TEST FACILITY

This facility incorporates the latest advancements in liquid metal handling, and utilizes advanced instrumentation for measurement of the operating and performance characteristics of the dynamic shaft seals. Alkali metal can be supplied to the test section over a range of conditions of pressure, temperature and flowrates.

Alkali metal seal testing can be performed at speeds in excess of 35,000 rpm and at temperatures ranging from about 150°F to 1400°F. Vacuum on the space side of the seals can be maintained at 10^{-6} mm Hg for long time endurance testing. Thus, outer space environmental testing of liquid metal dynamic seals can be evaluated in this facility.

Design Specifications:

Maximum operating temperature:	1400°F
Maximum operating pressure:	80 psig
Maximum flow rate:	4 gpm
Alkali metal containment material:	316 Stainless Steel

Equipment and Operation:

The test facility isometric screw is shown in Figure 15 and schematic screw is shown in Figure 16. Both the alkali metal loop and the argon gas loops are shown. Argon gas is primarily used as a cover gas over the alkali metal and as a lubricant for the gas bearings in the test rig. Because of the relatively large rate of argon consumption (36 to 50 scfm), it is necessary to reclaim argon for re-use. Reclaimation is accomplished by a system independent to the seal test facility and is therefore not shown on the enclosed Figures.

Liquid metal will be circulated by a EM Pump, which is capable of pumping 4 gpm at 80 psig. Liquid metal is pumped through a heat exchanger where the temperature is increased to within 200°F of that in the return line. After the heat exchanger, the alkali metal passes through a 15 micron filter. From the filter, the alkali metal flows through two parallel, electrically heated tubes and flow control valves to the test seal. From the test rig, the alkali metal (possibly containing a small amount of argon) is returned through the heat exchanger where it is cooled to within 200°F of the incoming liquid metal. Out of the heat exchanger, the liquid metal passes through a cooler (cold trap) where the metal is cooled by air to 200 - 300°F. The cooler is packed with wire mesh for trapping oxides at the lower temperature. From the cooler, the liquid metal enters a head tank where the entrapped argon is released from the alkali metal. The argon, containing some alkali metal vapor, is removed from the top of the head tank and is cooled to near room temperature before it enters a liquid nitrogen cooled vapor trap. Essentially, all traces of alkali metal are to be removed from the argon gas by the vapor trap and subsequent filter system. Argon from the vapor trap goes to the reclaimation system where it is further cleaned and repressurized.

The seal testing requires a vacuum of 10^{-6} mm Hg on one side of the seal. This vacuum will be obtained by a 5 1/4" I.D. diffusion pump in series with a 15 cfm mechanical pump. A liquid nitrogen cooled vapor trap and a liquid nitrogen cooled chevron baffle separate the test seal from the diffusion pump. These traps prevent mixing of alkali metal vapor with the diffusion pump oil vapor.

Progress

Detail design of all major loop components has been completed. Drawings and specifications for these components are now out for bids. Orders will be placed by January 31. Piping and loop layout drawings are about 90% complete. The electrical control and instrumentation systems have been specified and are now in the vendor's shop for fabrication. About 85% of standard hardware such valves, flowmeters, pressure gages, liquid level gages, vacuum pumps and gages, etc. have been received.

The outstanding engineering and drafting yet to be done involves service connections such as air, electrical and scrubber exhaust connections. This work has been started and is scheduled to be finished about the end of February. Drawings showing the installation of trace heating elements and thermal insulation are yet to be completed.

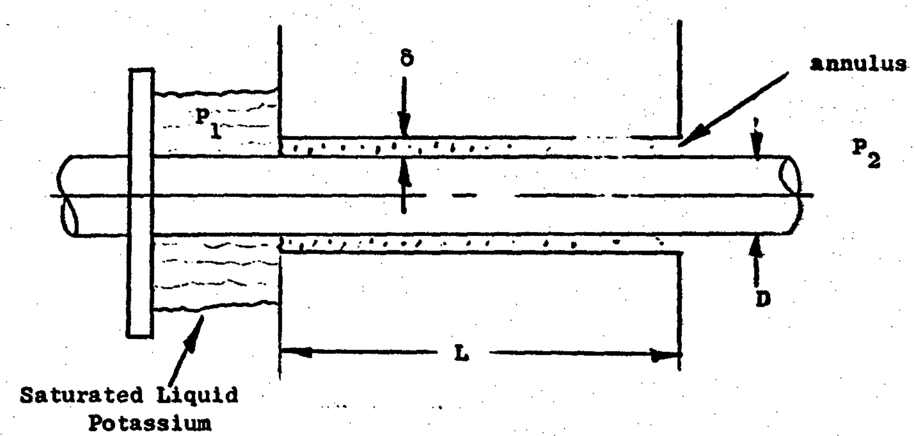
V. ANALYTICAL INVESTIGATION OF LEAKAGE FROM CLOSE GAP SEALS

A study has been made of molecular flow from close gap seals exhausting to vacuum. The effect of temperature and vapor pressure on leakage rates has been included in the analysis. The study predicts that a 2.0 inch long annulus is sufficient to restrict flow to 1.0 lb/year when potassium is in the temperature range 700 - 1000°R but at 1400 R such leakage rates can only be obtained with clearances less than .001 inches.

I. MOLECULAR FLOW FROM CLOSE GAP SEALS

A. Introduction

The basic problem is to determine the leakage rate of potassium vapor from a pool of saturated liquid potassium when the vapor passes through an annulus, as shown below. In the temperature range considered (700°R - 1400°R) the vapor density is too small to allow the vapor flow rate to be computed by the usual continuum equations.



The important dimensionless parameter which characterizes the flow regime is the Knudsen Number (K_n) which is the ratio of the molecular mean free path

(λ) to the clearance (δ). When the Knudsen Number is small, the flow is governed by the familiar continuum equations. When the Knudsen Number is large, the flow is governed by the "free-molecular" equations which assume that the flow resistance is due entirely to molecular collisions with the bounding walls since inter-molecular collisions are rare. In between these extremes lie the rather vaguely defined slip-flow and transition regimes. The mathematical distinction between these regimes will be discussed in greater detail in the section concerning leakage at 1400°R.

B. General Data

D = shaft dia = 1.0 inches

L = annulus length = 0.5 - 2.0 inches

δ = annulus clearance = 0.001 - 0.010 inches

T = potassium saturation temperature = 700°R - 1400°R

P_2 = ambient pressure at the atmospheric end of the annulus = 10^{-6} mm Hg

P_1 = saturation pressure of potassium

Flow is isothermal

C. Mean Free Path

The type of vapor flow regime (continuum, slip, free molecular) depends on the Knudsen Number

$$K_n = \frac{\lambda}{\delta} \qquad \text{(Ref. 1)}$$

where

λ = mean free path

δ = annulus clearance

The mean free path (λ) is predicted from kinetic theory to be

$$\lambda = \frac{3\,\nu}{\bar{V}} \qquad \text{(Ref. 1)}$$

where

$$\nu = \frac{\mu}{\rho} = \text{kinematic viscosity of the vapor} \qquad \text{(Ref. 2)}$$

$\overline{V}$ = mean molecular velocity

The mean velocity is calculated from

$$\overline{V} = \sqrt{\frac{8}{\pi \gamma}}\ V_s \qquad \text{(Ref. 1)}$$

where

γ = specific heat ratio

V_s = velocity of sound $= \sqrt{\gamma\ g_o\ R\ T}$

R = gas constant

T = absolute temperature

For saturated potassium vapor (Ref. 2)

T °R	$\overline{V}$ ft/sec	ν ft^2/hr	λ in
700	1518	12.03 (10^5)	7.93
800	1615	60.67 (10^3)	0.376
900	1712	62.00 (10^2)	0.036
1000	1804	100.0 (10)	0.0056

Using the largest clearance anticipated (0.010 inches), the smallest Knudsen Number is

$$K_n \geq 0.56$$

which means that all vapor flows to be considered are in the free molecular regime. (Ref. 4, also see section to follow on leakage at 1400°R)

D. Evaporation Rate Through an Orifice

The maximum evaporation rate from the surface of a liquid is predicted by kinetic theory to be

$$\frac{W_{max.\ evap.}}{A} = \frac{P_{sat}}{\sqrt{\frac{2\pi RT}{g_o}}}$$

(Ref. 3)

where

$\frac{W_{max.\ evap.}}{A}$ = maximum mass rate of evaporation per unit area

$P_{sat.}$ = saturation pressure

Actual evaporation rates are slightly less than the maximum predicted by theory. Experimental results given by Dushman (Ref. 4) are in good agreement with this approximate equation. Dushman's data is as follows:

T °R	P psia	$lb/ft^2 sec$
655	1.934 (10^{-7})	3.91 (10^{-7})
712	1.934 (10^{-6})	3.75 (10^{-6})
781	1.934 (10^{-5})	3.58 (10^{-5})
864	1.934 (10^{-4})	3.42 (10^{-4})
968	1.934 (10^{-3})	3.22 (10^{-3})
1100	1.934 (10^{-2})	3.03 (10^{-2})

Since these values represent the evaporation rates from a surface (which behaves like an orifice in a thin plate), they are the evaporative losses of the lubrication case of an annulus of zero length (L).

E. Evaporation Rate Through An Annulus

When the interface between the liquid and vapor occurs within an annulus, the walls of the annulus restrict the molecular flow and reduce the evaporative rate below the maximum orifice rate.

The free molecular flow rate through an annulus is given by

$$W = \frac{2}{3}\ \frac{F\bar{V}A}{RT}\ (P_1 - P_2) \qquad \text{(Ref. 4)}$$

where

$$W = \text{molecular mass flow rate in the annulus}$$

$$P_1, P_2 = \text{pressures at the ends of the annulus}$$

$$F = \text{a function of the annulus geometry (Ref. 4)}$$

$$= \frac{\delta}{L}\left[\frac{1}{1 + \frac{8}{3} K \frac{\delta}{L}}\right], \quad \frac{L}{\delta} > 1.5 \qquad \text{(See Fig. 2)}$$

$$K = \text{tabulated function of } \left(\frac{L}{\delta}\right) \qquad \text{(Ref. 5)}$$

$$A = \pi D \delta = \text{cross-section area of the annulus}$$

The evaporation rate from the interface is controlled by the resistance of the annulus to vapor flow. This is expressed mathematically through the continuity equation as follows:

$$W_{evap} = \frac{2}{3} \frac{F\bar{V}A}{RT} (P_1 - P_2)$$

where

$$P_1 = \text{saturated vapor pressure}$$

$$P_2 = \text{ambient pressure}$$

Since the lowest vapor pressure (P_1 at 700°R) is 1.1989 (10^{-6}) psia and $P_2 = 10^{-6}$ mm Hg = 1.934 (10^{-8}) psia the maximum evaporative rate through the annulus is approximated by

$$W_{evap.\ max.} \cong \frac{2}{3} F\bar{V}A \rho_{vapor}$$

where

$$\rho\ vapor = \text{saturated vapor density}$$

Assuming that the maximum allowable leakage rate is of the order of 1 lb/ year the following table indicates the length annulus required to maintain this as a maximum rate:

$T = 700°R$

δ inches	$\frac{2}{3}\rho_{vap}\overline{V}A$	L inches	$W_{max.\ evap.}$ lb/yr (Ref. 4)
0.001	43.4 (10^{-4})	0.000	16.93 (10^{-4})
0.005	217 (10^{-4})	0.000	84.64 (10^{-4})
0.010	434 (10^{-4})	0.000	169.3 (10^{-4})

$T = 800°R$

δ inches	$\frac{2}{3}\rho_{vap}\overline{V}A$	L inches	$W_{max.\ evap.}$ lb/yr (Ref. 4)
0.001	1100 (10^{-4})	0.000	(4.52) (10^{-2})
0.005	5500 (10^{-4})	0.000	(22.6) (10^{-2})
0.010	11000 (10^{-4})	0.000	(45.2) (10^{-2})

$T = 900°R$

δ inches	$\frac{2}{3}\rho_{vap}\overline{V}A$	$\frac{L}{\delta}$	L inches	$W_{evap.\ max.}$ lb/yr
0.001	1.342	0	0.000	0.5
0.005	6.71	5.8	0.029	1.0
0.010	13.42	13.42	0.134	1.0

$T = 1000°R$

δ inches	$\frac{2}{3}\rho_{vap}\overline{V}A$	$\frac{L}{\delta}$	L inches	$W_{evap.\ max.}$ lb/yr
0.001	9.84	8.8	0.009	1.0
0.005	49.2	49.2	0.246	1.0
0.010	98.2	98.2	0.982	1.0

Flow at 1400°R

Calculating the evaporation rate at 1400°R presents some special problems since the mean free path is

$$\lambda = 3 \frac{\nu}{\bar{V}} = 68.7 \ (10^{-6}) \text{ inches}$$

and the Knudsen Number range is

δ inches	K_n	Regime
0.001	0.0687	free-molecular, transition
0.005	0.01370	transition, slip
0.010	0.00687	slip

In the slip-flow regime the flow rate is given by (Ref. 3)

$$W = \frac{\pi D}{24} \left(\frac{\delta^3}{\mu RT}\right) \left(\frac{P_1^{\,2} - P_2^{\,2}}{L}\right) \left[1 + 6 \ \frac{2 - f}{f} \ K_n\right]$$

where

f = specular reflection coefficient

Since $P_1^{\,2} >> P_2^{\,2}$, using $\nu = \frac{1}{3} \lambda \sqrt{\frac{8 g_o RT}{\pi}}$

the flow rate may be approximated by

$$W \cong \frac{\pi}{64} \ \frac{-\delta}{L} \left[\frac{1}{Kn} + 6 \left(\frac{2 - f}{f}\right)\right] \ \frac{\rho_1 \bar{V} A}{RT}$$

For Knudsen Numbers in the free-molecular flow regime this equation reduces to

$$W \cong \frac{\pi}{64} \ \frac{\delta}{L} \left[6 \ \frac{2 - f}{f}\right] \frac{\rho_1 \bar{V} A}{RT}$$

When $\frac{\delta}{L} << 1.0$, f may be evaluated by comparing this equation with the free molecular flow equations used previously, so that

$$\frac{2}{3} = \frac{\pi}{64}\left[6\left(\frac{2-f}{f}\right)\right]$$

or

$$6\left(\frac{2-f}{f}\right) = \frac{128}{3\pi} = 13.56 \qquad \text{(Ref. 6)}$$

The approximate equation which covers all flow regimes is then

$$W = \frac{\pi D}{24}\left(\frac{\delta^3}{\mu RT}\right)\left(\frac{P_1^{\,2} - P_2^{\,2}}{L}\right)\left[1 + 13.56\, K_n\right]$$

when

$$\frac{\delta}{L} << 1.0$$

Note that if $K_n = 1/2$, the error introduced by neglecting the plane Poiseuille flow contribution, as in the calculations at lower temperatures, amounts to

$$\frac{1}{1 + 13.56\,(1/2)} = 12.86\%$$

Using the approximate equation above at 1400°R for $W_{evap.\ max.} =$ 1.0 lb/year

δ inches	K_n	L inches
0.001	0.0687	3.15
0.005	0.01374	241.0
0.010	0.00687	1780.0

F. Conclusion

It appears that a 2.0 inch long annulus is sufficient to restrict the flow to 1.0 lb/year when the potassium is in the temperature range 700-1000°R but at 1400°R such leakage rates can be obtained only when δ = 0.001 inches.

SYMBOLS

1. **Slinger Seal**

R	Reynolds number, taken at the disk tip, dimensionless, $\frac{\omega a^2}{\nu}$.
ΔC_m	Torque coefficient for disk partially wetted on two sides, dimensionless, $\frac{2M}{\rho\,\omega^2\,a^5}$.
P_H	Static pressure being sealed by the water, psig.
$P_{Dynamic\ L}$	Dynamic pressure of sealing fluid, water, on the low pressure side of the disk, psig.
a	Radius of stationary disk, ft.
r_L	Radius of fluid from the center of the disk on the low pressure side, ft.
r_H	Radius of fluid from the center of the disk on the high pressure side, ft.
ν	Kinematic viscosity, ft^2 per second.
ρ	Sealing fluid mass density, lb sec^2/ft^4.
k	Ratio of angular velocities, dimensionless, $\frac{\beta}{\omega}$.
ω	Angular velocity of rotating housing, radians per second.
N	Rotating housing velocity, rpm.
β	Angular velocity of sealing fluid, water, radians per second.

SYMBOLS (continued)

M	Frictional torque (moment), ft. lb.
$\frac{a - r}{a}$	Radius Ratio, dimensionless.
HP	Horsepower.
W	Cooling flow, gpm water.
S_L	Spacing between disk and rotating housing on the low pressure side of the disk, ft.
η	Seal Efficiency.
F_N	Thrust.

2. Screw Seal

D	Shaft diameter.
e	Flight width.
h	Depth of groove in shaft.
K_1 , K_2	Constants in sealing equation.
L	Threaded length of shaft.
n	Number of threads.
N	Shaft angular velocity, rpm.
ΔP	Pressure drop across seal.

SYMBOLS (continued)

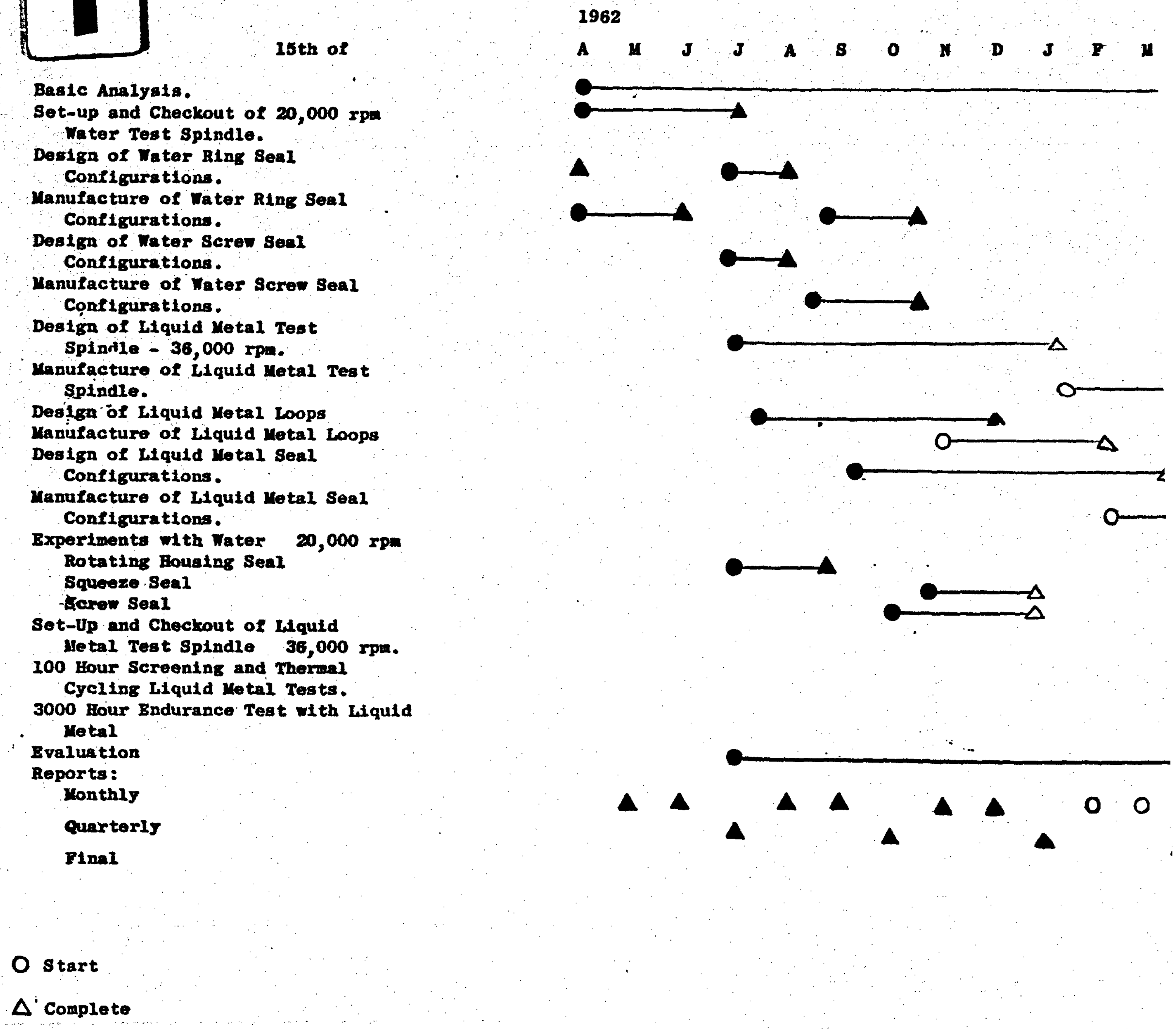
Re	Reynolds number
t	Thread pitch $= \pi D \tan \phi$
t^*	$t^* = t/n$
V	Shaft Speed.
w	Width of thread channel.
w'	Width of thread land.
α	h/w
β	δ/h
γ	$w/(w + w')$
δ	Radial clearance.
μ	Absolute viscosity.
ν	Kinematic viscosity.
φ	Thread helix angle

Subscripts

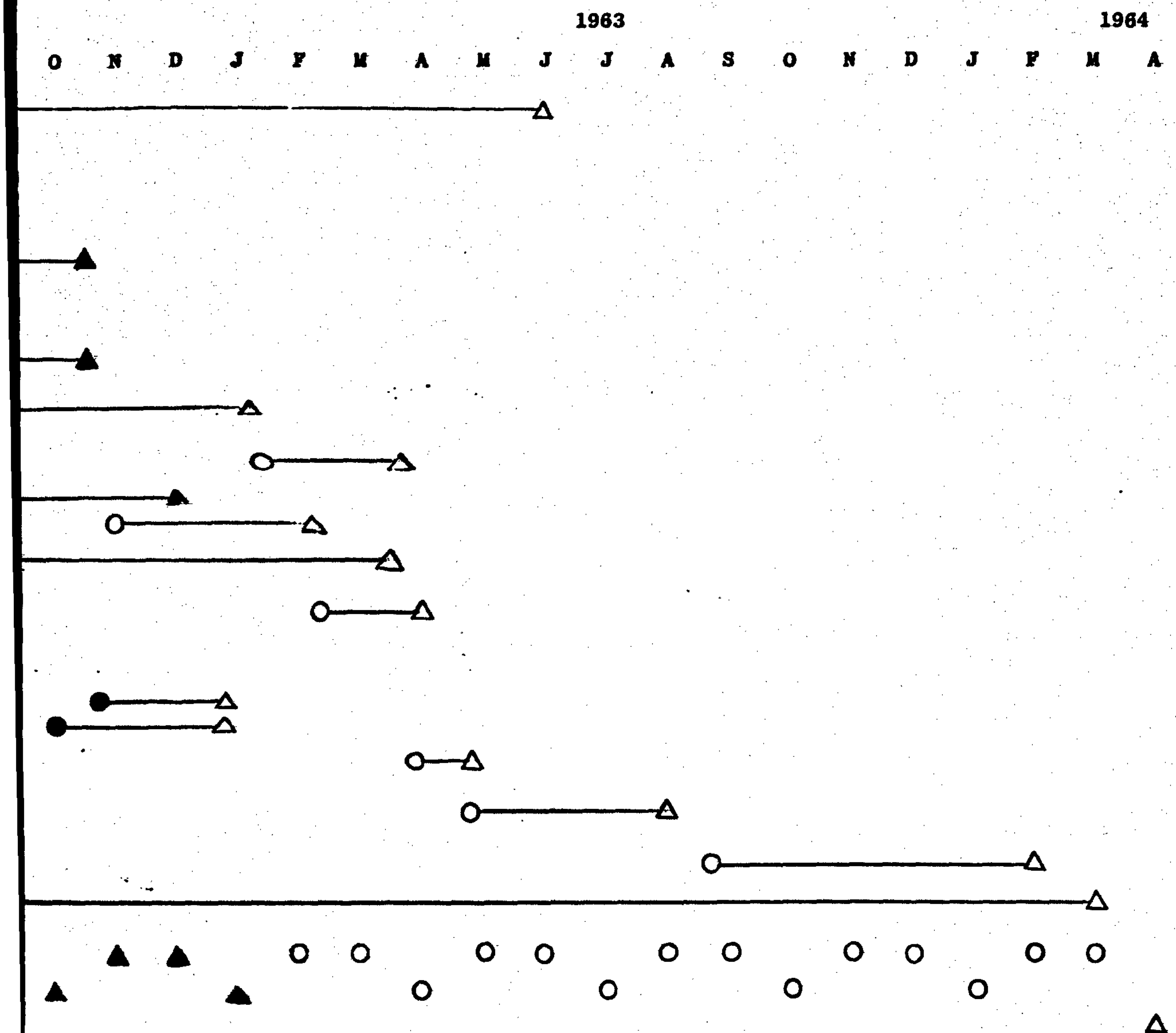
cr	Critical
opt	Optimum value.

1

Figure 1. Dynamic Seal Work Schedule



Dynamic Seal Work Schedule



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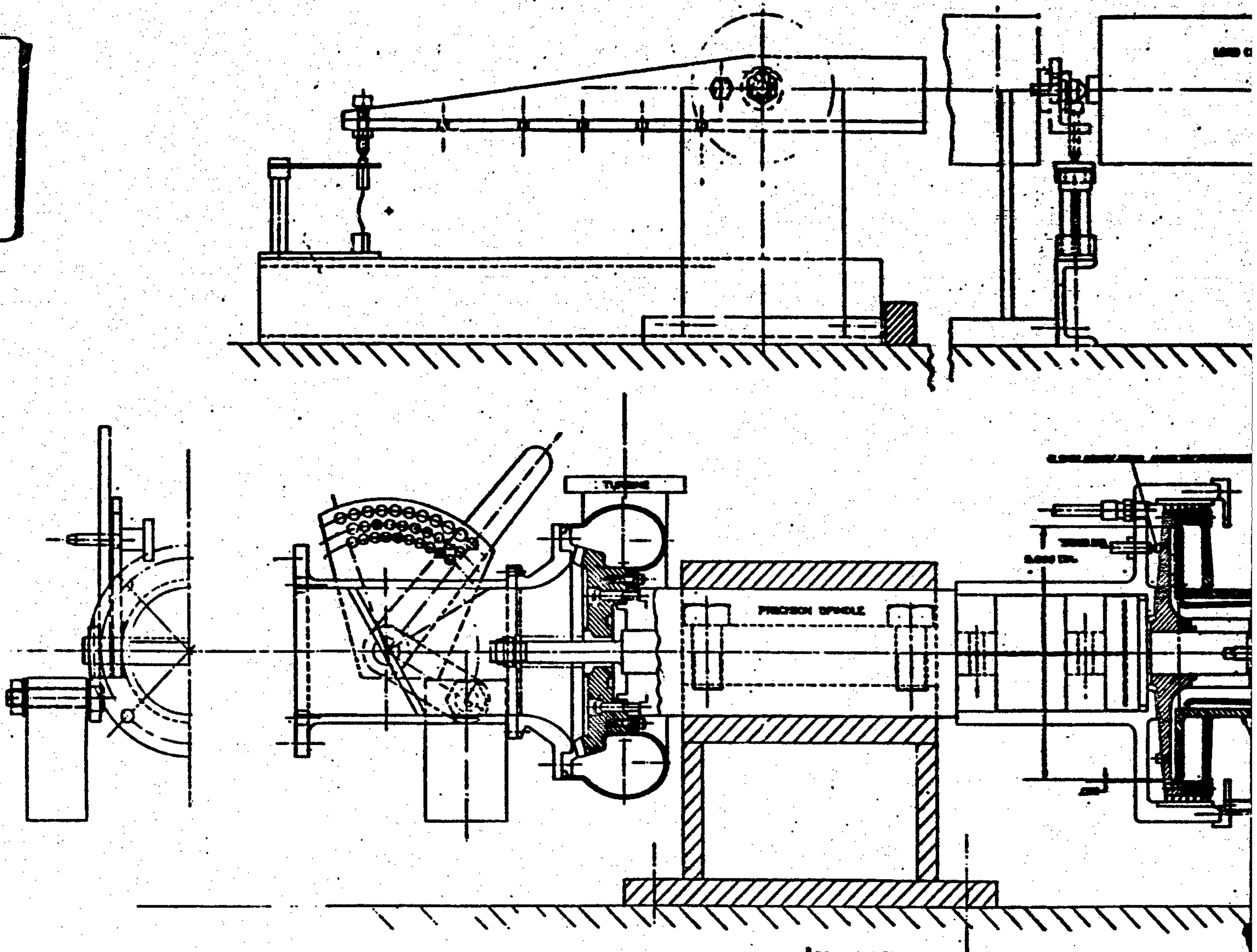


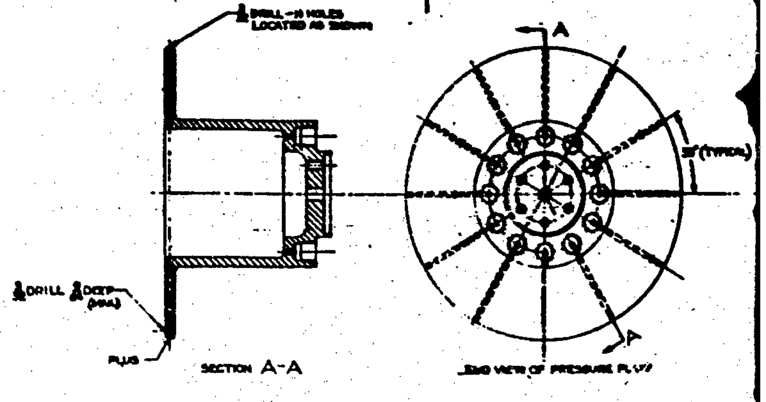
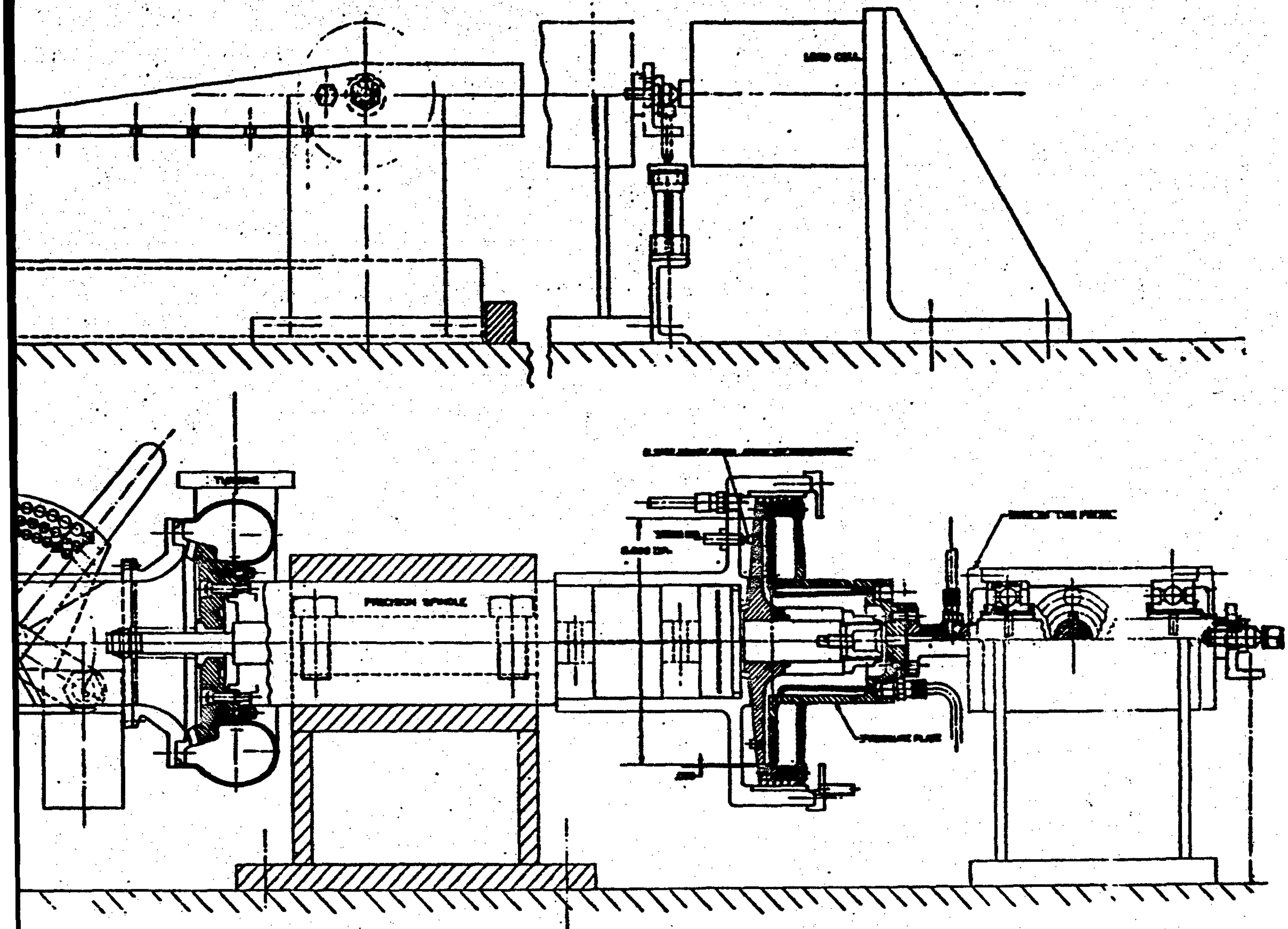
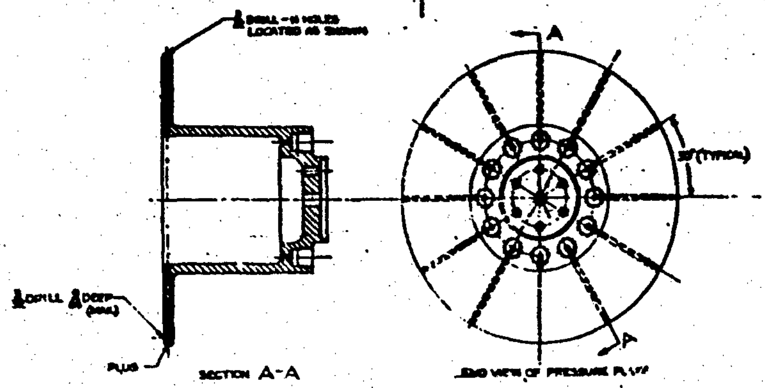


Figure 2. High Speed Dynamic Seal Test Rig (Working Fluid Water)





Seal Test Rig
ter)

2

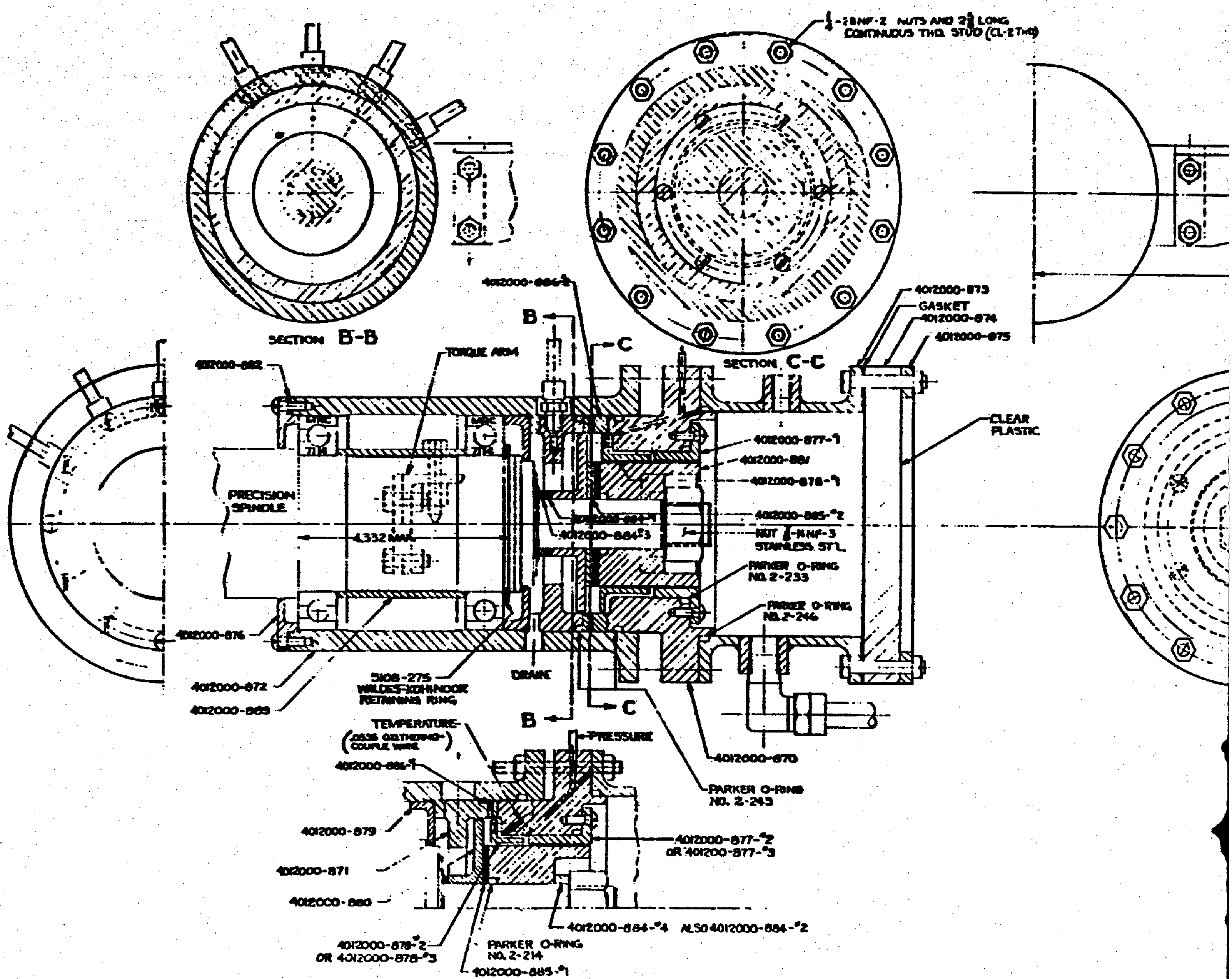


Figure 3. Seal Test Machine - Design No. 2

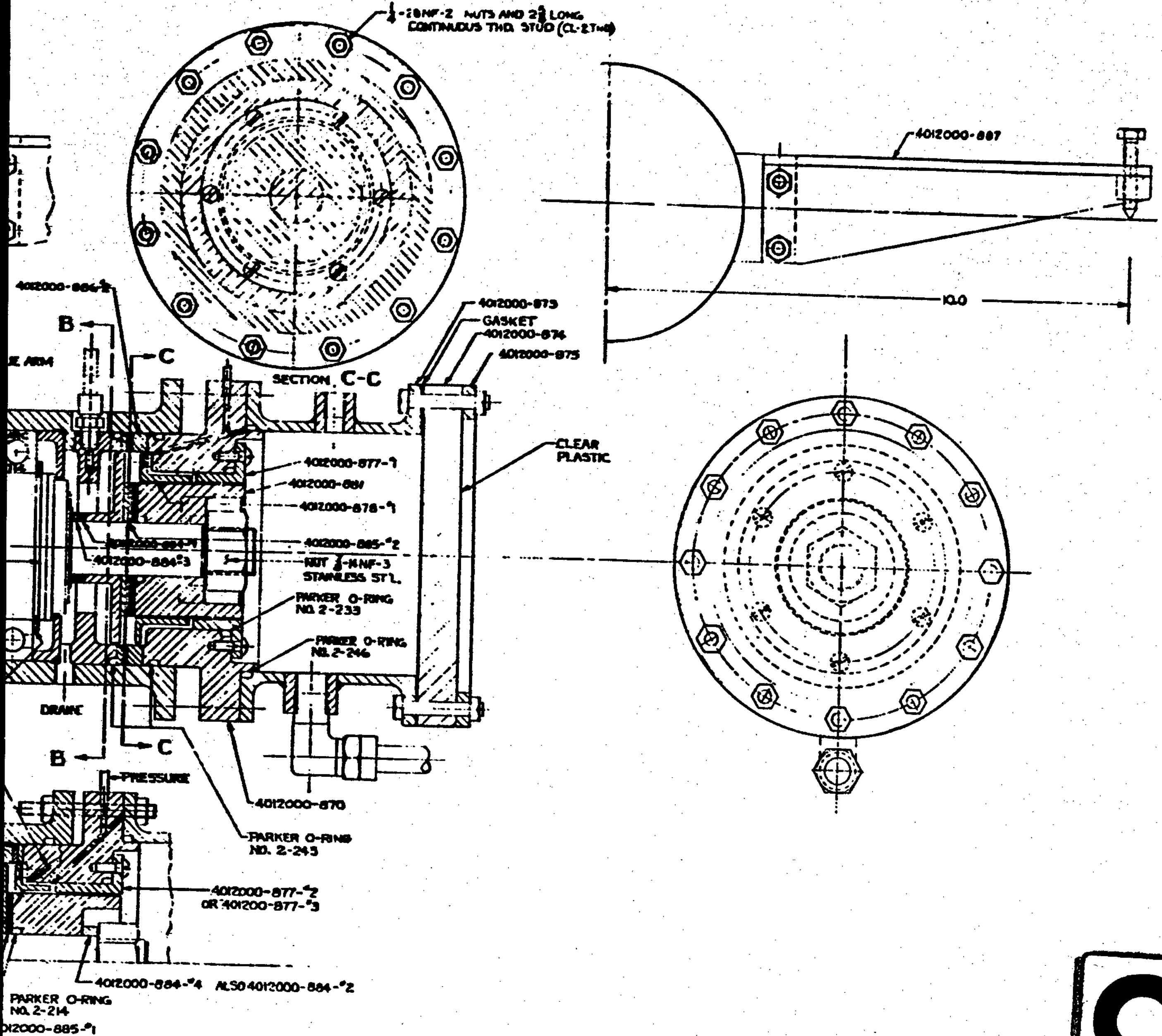


Figure 3. Seal Test Machine - Design No. 2

2

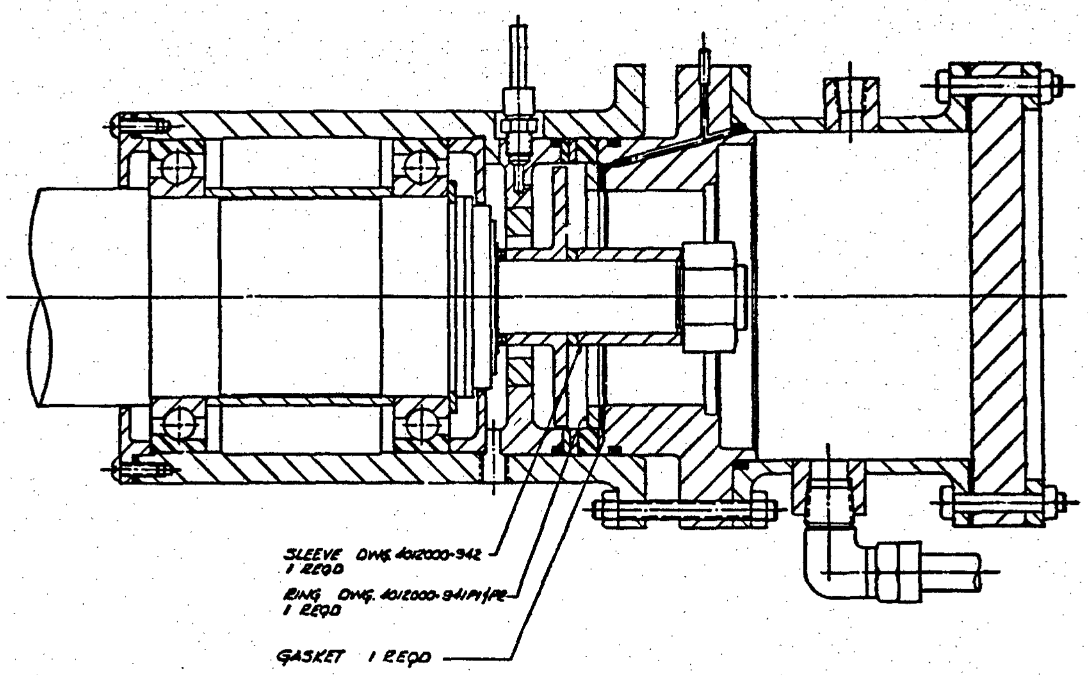


Figure 4. Rotating Disk Seal Configuration.

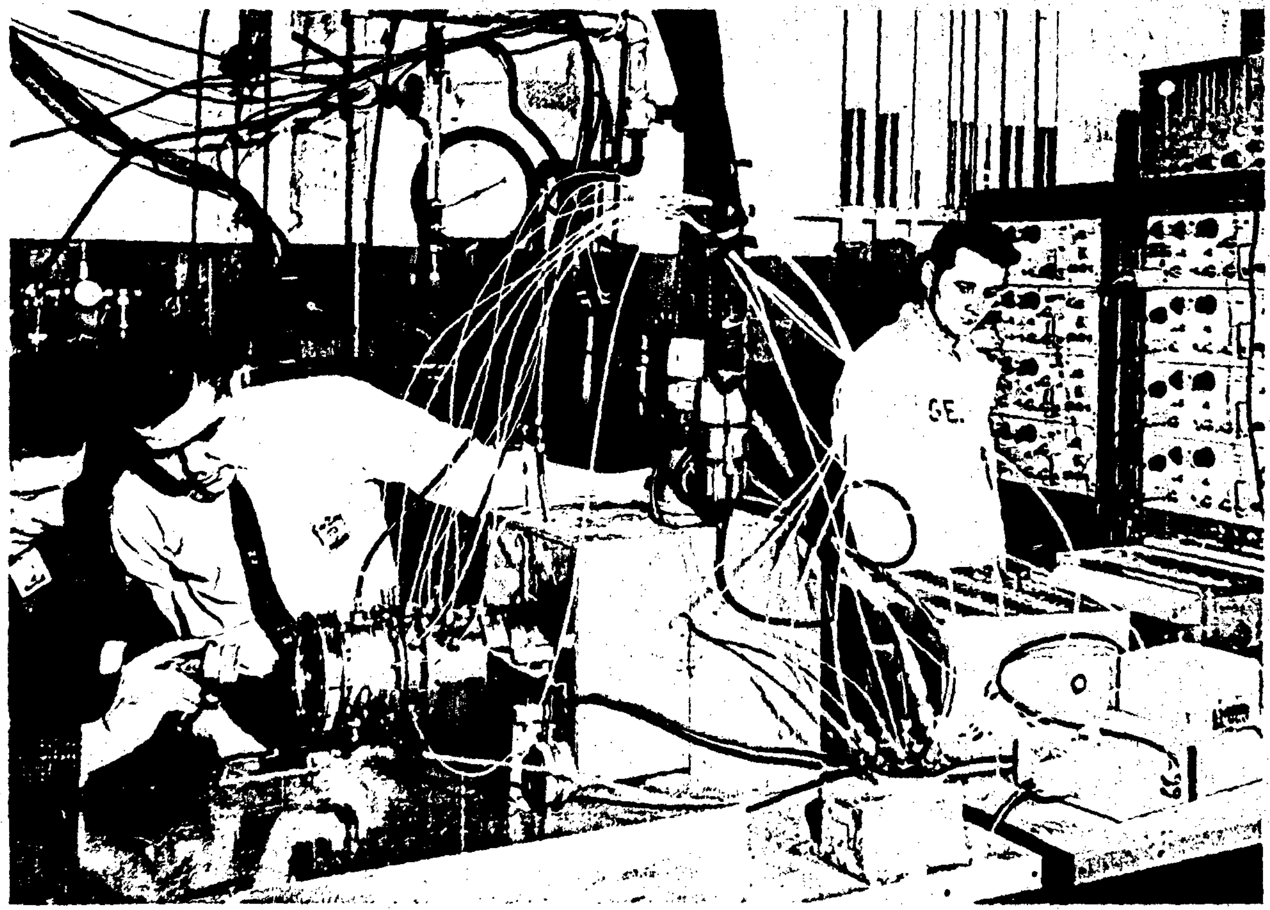


Figure 5. Photograph of Rotating Disk-Squeeze Seal Test Rig in Operation, No. C212964.

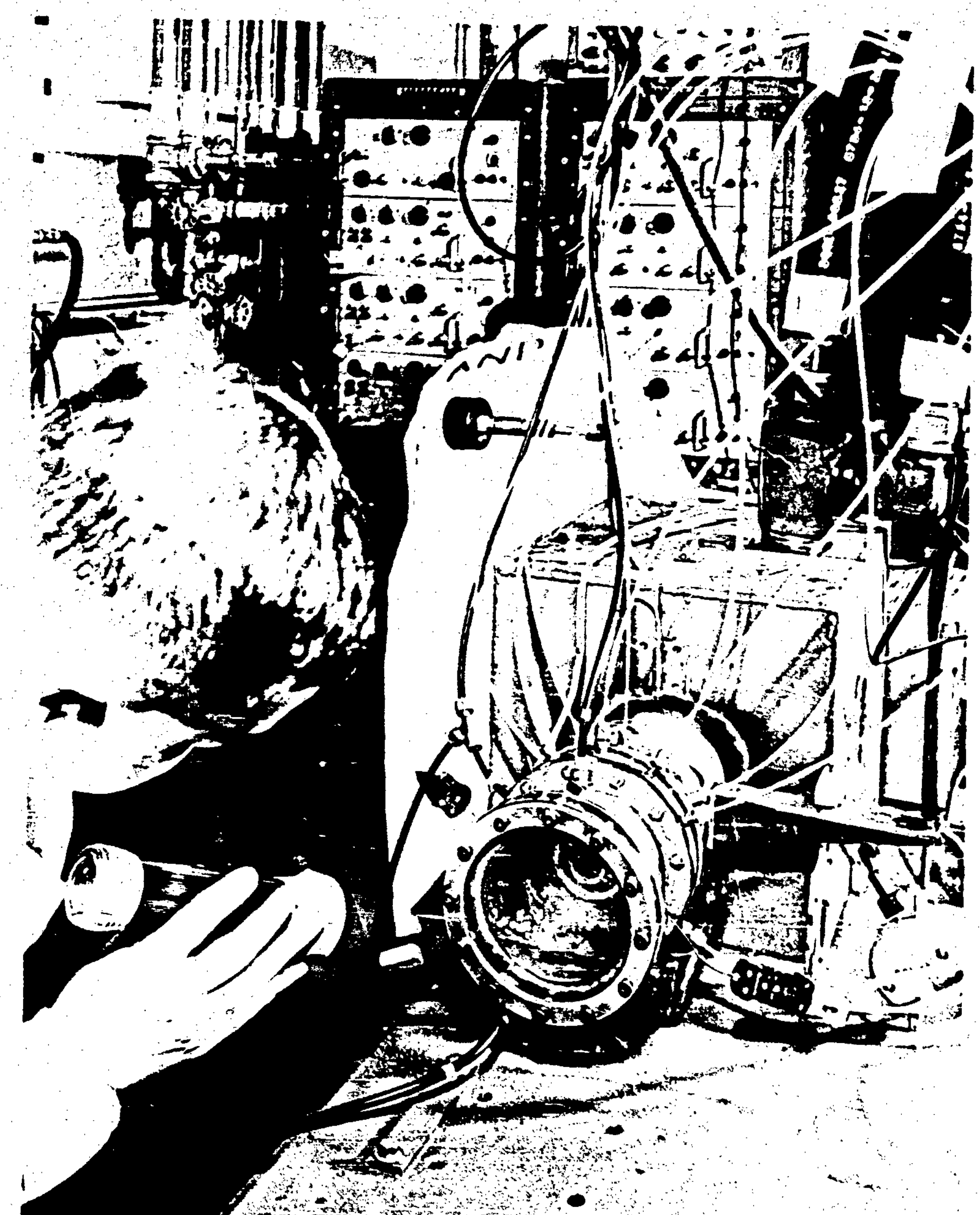


Figure 6. Photograph Test Rig in Operation, No. C212965.

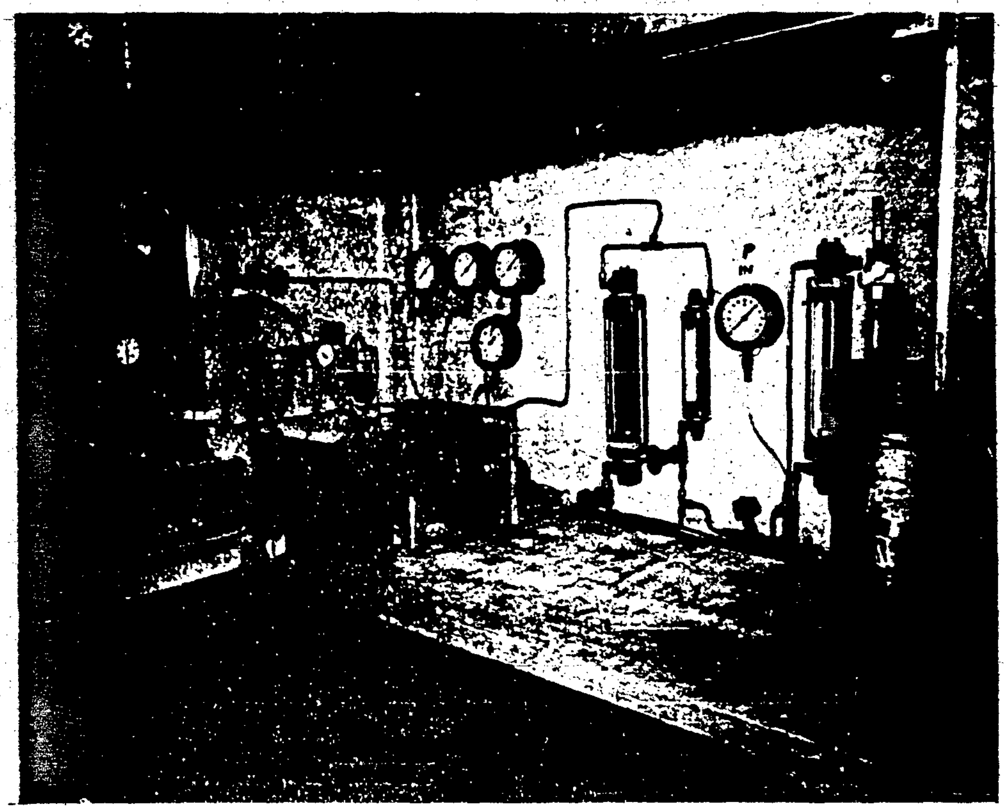


Figure 8. Picture of Test Rig

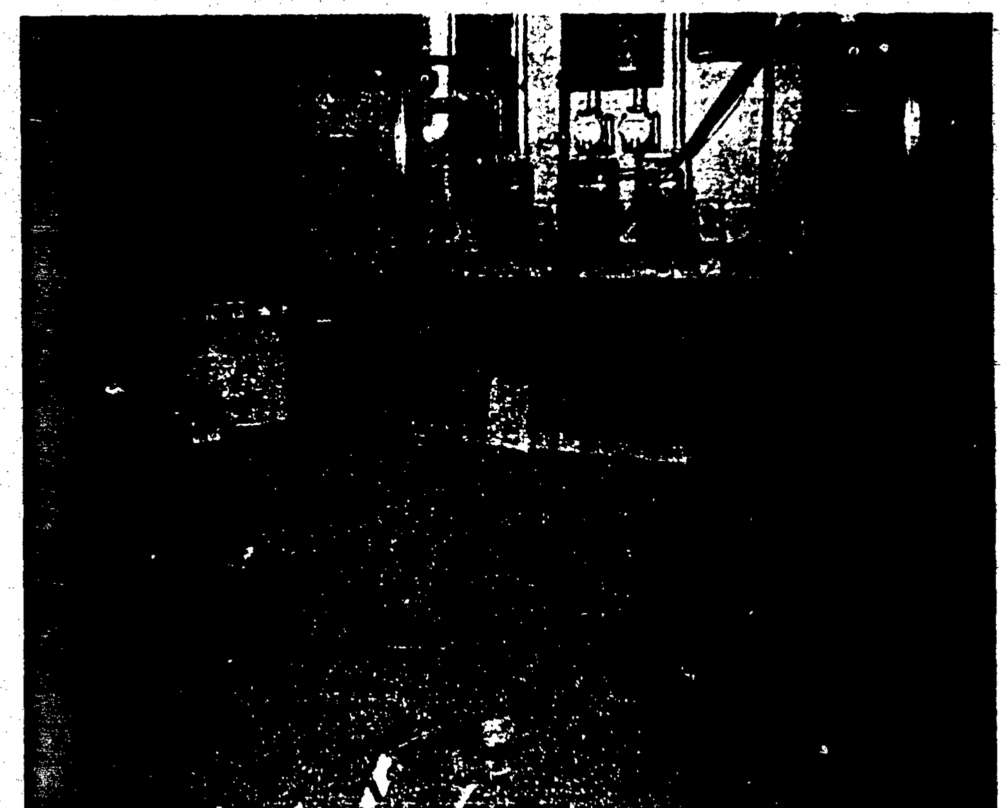


Figure 8 Picture of Instrumentation

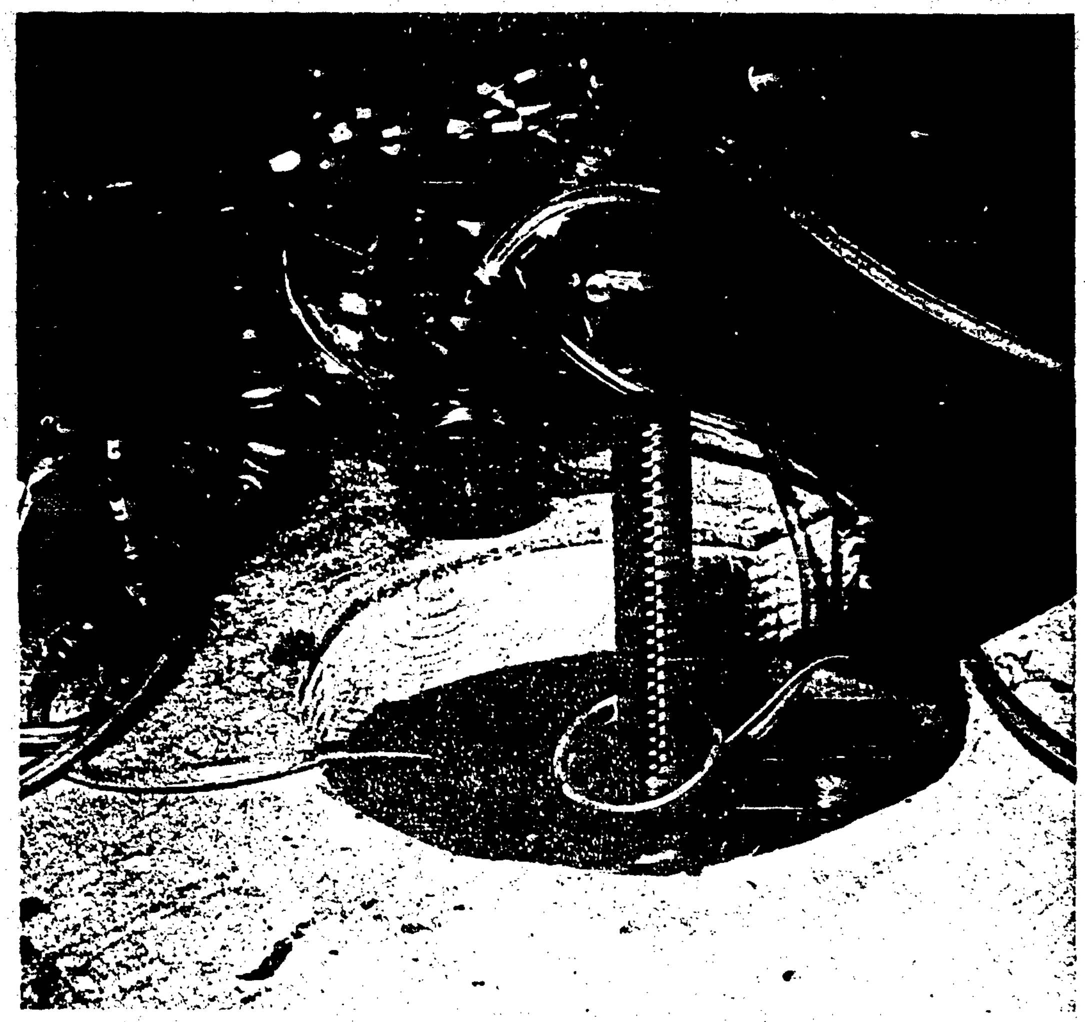


Figure [illegible]. Picture of Screw

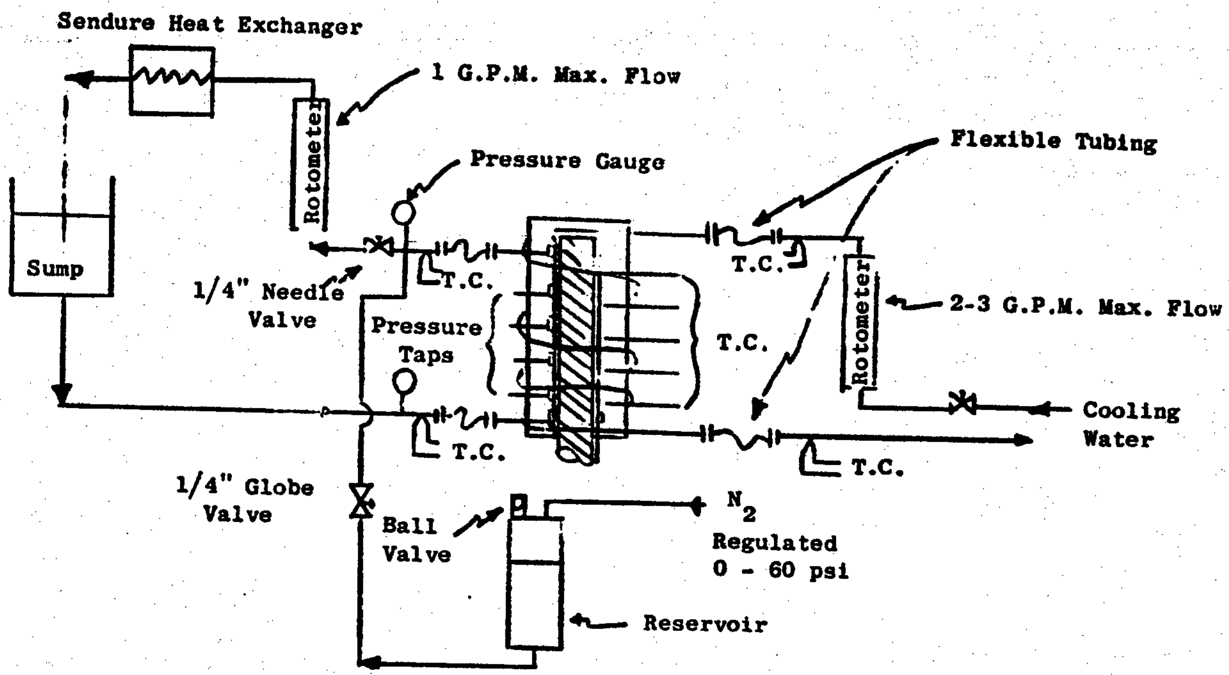


Figure 12. Test Loop and Instrumentation for Screw Seal Test.

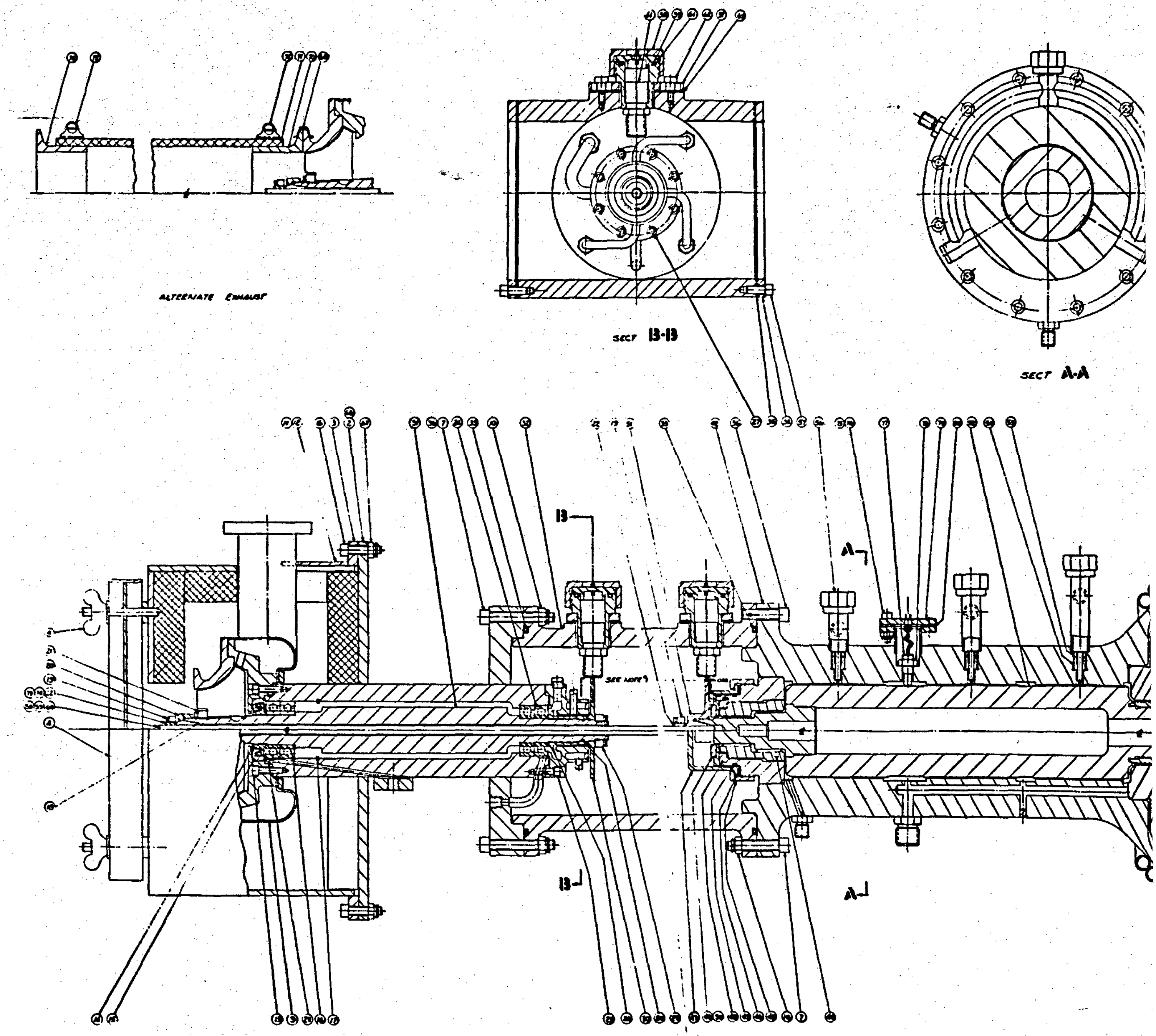
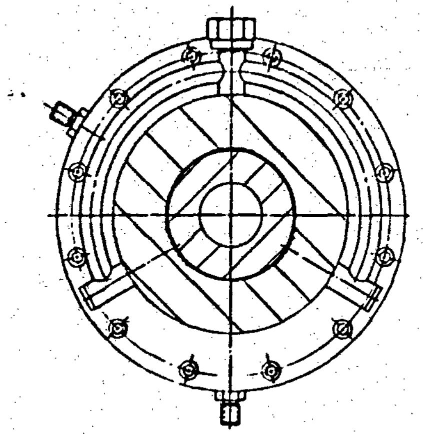
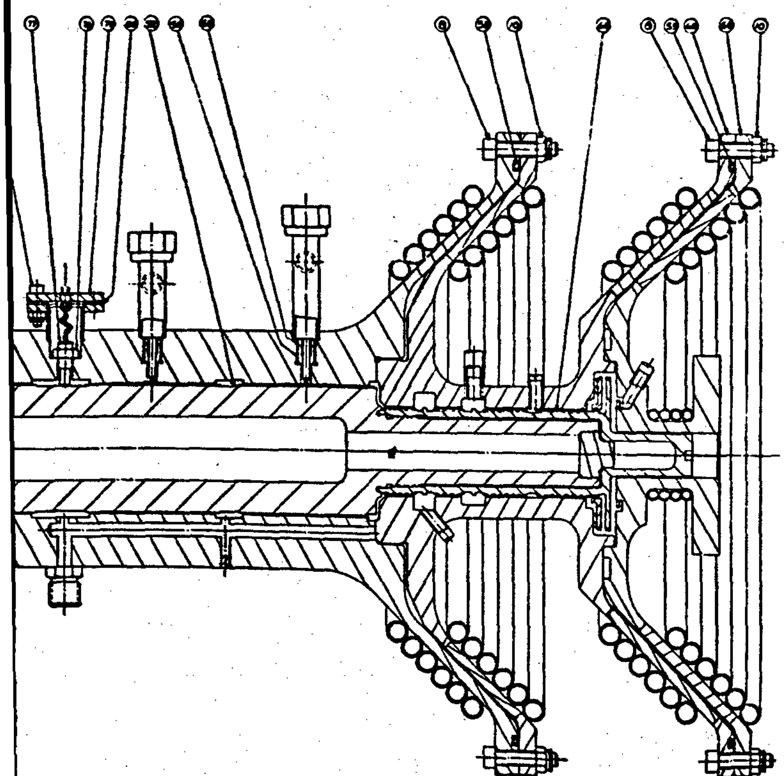


Figure 13



SECT A-A



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	1		[illegible]		
	2	[illegible]	BRG HOUSING	E13	1
	3	[illegible]	SILENCER	E13	1
	4	[illegible]	BAFFLE	C14	1
	5		[illegible]	C14	4
C	6		[illegible]	E19	6
C	7		[illegible]	E8	19
C	8		[illegible]	E8	49
	9	[illegible]	[illegible]	A15	7
C	10	[illegible]	NUT	E8	33
	11	[illegible]	COVER	E13	1
C	12		[illegible]	E19	4
C	13		[illegible]	A13	6
	14	[illegible]	ROTATING SEAL	A14	1
	15	[illegible]	STATIONARY SEAL	A14	1
D	16	[illegible]	BALL BEARING	A19	2
E	17	[illegible]	RETAINING RING	A19	1
	18		O-RING [illegible]	B14	1
C	19	[illegible]	NUT	C14	1
	20	[illegible]	NUT	C14	1
C	21	[illegible]	NUT	C14	1
	22	[illegible]	COLLET	C14	2
F	23	[illegible]	O-RING	A19	1
D	24	[illegible]	BALL BEARING	E8	2
	25	[illegible]	STATIONARY SEAL	A11	1
	26	[illegible]	ROTATING SEAL	A11	1
C	27		[illegible]	A11	6
F	28	[illegible]	O-RING	A11	1
C	29	[illegible]	NUT	A11	1
	30	[illegible]	SERRATED DISK	A11	1
	31	[illegible]	SHAFT	F12	1
	32	[illegible]	INSTRUMENTATION SECTION	E12	1
G	33	[illegible]	O-RING	E11	2
	34	[illegible]	COVER	E11	2
	35	[illegible]	GASKET	E11	2
L	36		[illegible]	E11	3
	37	[illegible]	[illegible] HOLDER	H11	2
	38	[illegible]	NUT	H11	2
F	39	[illegible]	O-RING	H11	2
	40	[illegible]	GASKET	H11	2
H	41	[illegible]	MAGNETIC PICKUP	H11	2
	42	[illegible]	BRG HOUSING	E11	1
	43	[illegible]	SEAL HOUSING	A11	1
	44	[illegible]	STATIONARY SEAL	A10	1
F	45	[illegible]	O-RING	A11	1
F	46	[illegible]	O-RING	A11	1
	47	[illegible]	[illegible]	A11	1
	48	[illegible]	NUT	A11	1
	49	[illegible]	THRUST WASHER	A11	1
	50		[illegible]	A11	6
	51	[illegible]	SERRATED DISK	E11	
	52	[illegible]	SHAFT	E9	1
G	53	[illegible]	O-RING	E8	2
	54	[illegible]	[illegible]	E2	2
J	55	[illegible]	[illegible]	E9	9
K	56	[illegible]	CONNECTOR	E9	9
C	57		[illegible]	E10	15
	58	[illegible]	DUMMY SHAFT	C4	
	59	[illegible]	DUMMY SHAFT	C14	
	60	[illegible]	DUMMY SHAFT	C4	1
	61	[illegible]	SEAL PLATE	H11	2
	62	[illegible]	SHOULDER BOLT	H11	4
	63	[illegible]	SLEEVE	E8	
	64	[illegible]	SEAL	E7	1
	65	[illegible]	[illegible]	E7	
M	66	[illegible]	[illegible]	–	1
C	67	[illegible]	NUT	E3	8
	68	[illegible]	[illegible]	[illegible]	1
N	69	[illegible]	[illegible]	[illegible]	2
	72	[illegible]	SLEEVE ADAPTER	[illegible]	2
	74	[illegible]	[illegible]	[illegible]	
	75	[illegible]	HOSE CLAMP [illegible]	[illegible]	
	73	[illegible]	COLLET	C4	2
	76	[illegible]	COLLET	C4	2
C	75		[illegible]	E2	3
C	76	[illegible]	NUT	E10	8
P	77	[illegible]	SENSING HEAD	E10	2
	78	CONN	JAM NUT [illegible]	E10	2
	79	[illegible]	TERMINAL MTG PLATE	E10	2
	80	[illegible]	GASKET	E10	2

C-STANDARD PRESSED STEEL CO. JENKINTOWN, PA OR EQUIV.
D-MARLIN ROCKWELL CORP. JAMESTOWN, N.Y. OR EQUIV.
E-WALDES-KOHINOOR LONG ISLAND, N.Y. OR EQUIV.
F-COMPOUND N3407(S.N.A.N) PARKER SEAL CO. OR EQUIV.
G-UNITED AIRCRAFT PRODUCTS INC. DAYTON, OHIO OR EQUIV.
H-ELECTRO-PRODUCTS LABORATORIES CHICAGO, ILL.
J-DRILL & [illegible] DIA HOLE THRU PART [illegible]
K-CRAWFORD FITTING CO. CLEVELAND, OHIO OR EQUIV.
L-UNBRAKO WITH NYLOK INSERT STANDARD PRESSED STEEL CO. JENKINTOWN, PA OR EQUIV.
M-TO BE USED AS ALIGNMENT FIXTURE DURING ASSY.
N-MARMAN DIVISION AEROQUIP CORP. LOS ANGELS, CALIF OR EQUIV.
P-BENTLY NEVADA CORP. MINDEN, NEVADA OR EQUIV.

1. THIS DIMENSION MUST BE HELD WHEN ASSEMBLING [illegible] (ZONE C11)

ASSY-SEAL TEST SPINDLE

Figure 13

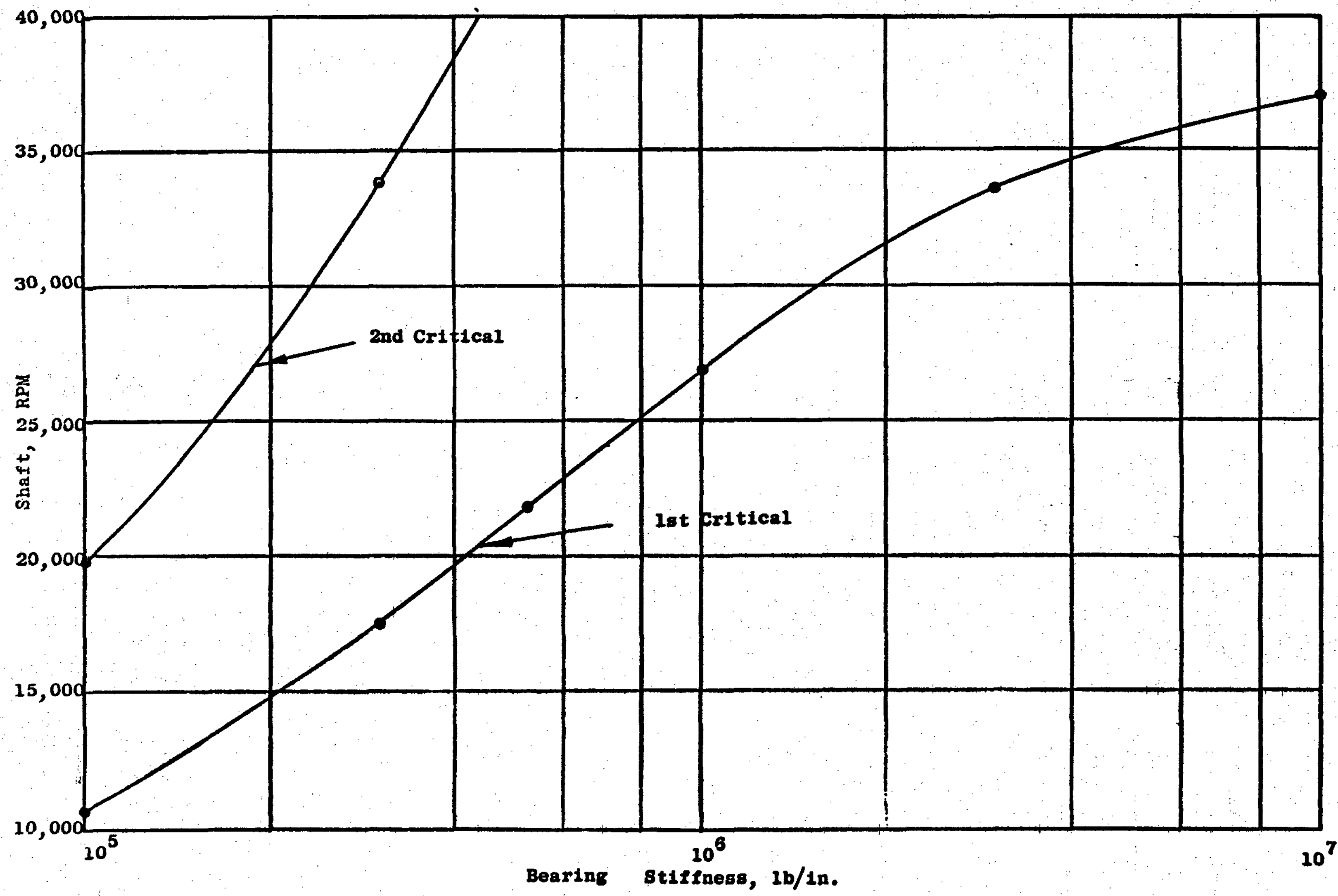
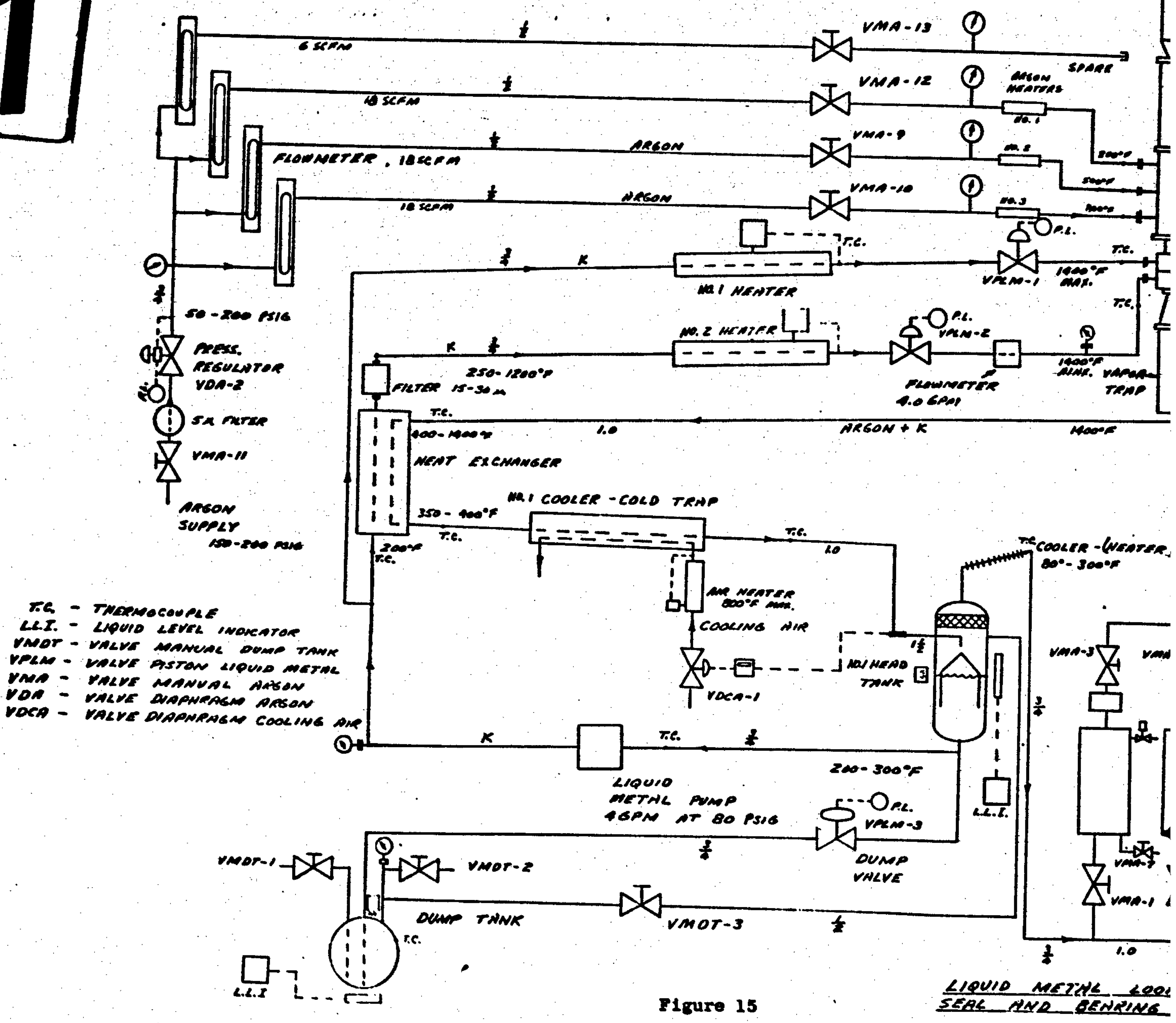
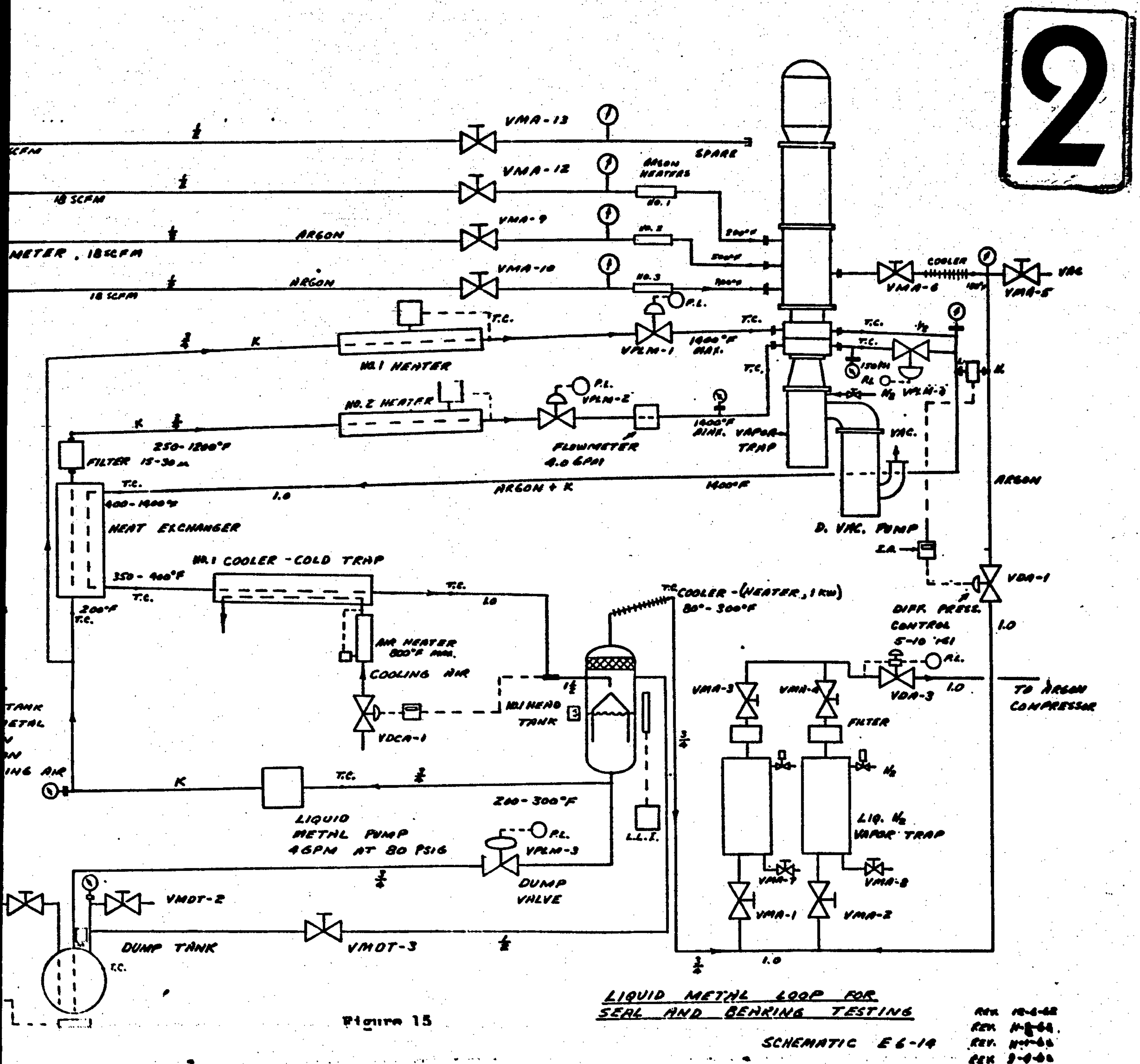


Figure 14. Plot of Shaft Critical Speeds for LM Test Spindle



VMA-13
6 SCFM
SPARE
VMA-12
18 SCFM
ARGON HEATERS
NO. 1
VMA-9
FLOWMETER, 18SCFM
ARGON
NO. 2
VMA-10
18 SCFM
ARGON
NO. 3
P.L.
T.C.
K
NO.1 HEATER
VPLM-1
1400°F MAX.
50-200 PSIG
PRESS. REGULATOR VDA-2
NO. 2 HEATER
P.L.
VPLM-2
250-1200°F
FILTER 15-30 μ
FLOWMETER 4.0 GPM
1400°F MAX.
VAPOR TRAP
5 μ FILTER
VMA-11
400-1400°F
1.0
ARGON + K
1400°F
HEAT EXCHANGER
ARGON SUPPLY 150-200 PSIG
NO. 1 COOLER - COLD TRAP
350-400°F
200°F
1.0
COOLER - (HEATER)
80°-300°F
AIR HEATER 800°F MAX.
COOLING AIR
T.C. - THERMOCOUPLE
L.L.I. - LIQUID LEVEL INDICATOR
VMDT - VALVE MANUAL DUMP TANK
VPLM - VALVE PISTON LIQUID METAL
VMA - VALVE MANUAL ARGON
VDA - VALVE DIAPHRAGM ARGON
VDCA - VALVE DIAPHRAGM COOLING AIR
VDCA-1
NO. 1 HEAD TANK
VMA-3
LIQUID METAL PUMP 4GPM AT 80 PSIG
200-300°F
P.L.
VPLM-3
DUMP VALVE
L.L.I.
VMDT-1
VMDT-2
DUMP TANK
VMDT-3
VMA-7
VMA-1
1.0
L.L.I
LIQUID METAL LOO
SEAL AND BEARING
SCHEM

Figure 15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VMA-13
SPARE
VMA-12
ARGON HEATERS
NO. 1
18 SCFM
VMA-9
ARGON
NO. 2
METER, 18 SCFM
VMA-10
ARGON
NO. 3
18 SCFM
P.L.
T.C.
K
NO. 1 HEATER
VPLM-1
1600°F MAX.
NO. 2 HEATER
VPLM-2
FLOWMETER
4.0 GPM
1600°F AIMS.
VAPOR TRAP
VAC.
VMA-6
COOLER
VMA-5
VAC
250-1200°F
FILTER 15-30 μ
ARGON + K
1600°F
ARGON
400-1600°F
HEAT EXCHANGER
D. VAC. PUMP
NO. 1 COOLER - COLD TRAP
350-400°F
200°F
COOLER - (HEATER, 1 KW)
80°-300°F
VDA-1
DIFF. PRESS. CONTROL
5-10 PSI
AIR HEATER
800°F MAX.
COOLING AIR
VDCA-1
NO. 1 HEAD TANK
VMA-3
VMA-4
FILTER
VDA-3
TO ARGON COMPRESSOR
LIQUID METAL PUMP
4 GPM AT 80 PSIG
200-300°F
L.L.I.
LIQ. N_2 VAPOR TRAP
VPLM-3
DUMP VALVE
VMOT-2
VMA-7
VMA-8
VMA-1
VMA-2
DUMP TANK
VMOT-3
1.0
LIQUID METAL LOOP FOR SEAL AND BEARING TESTING
SCHEMATIC E6-14

Figure 15

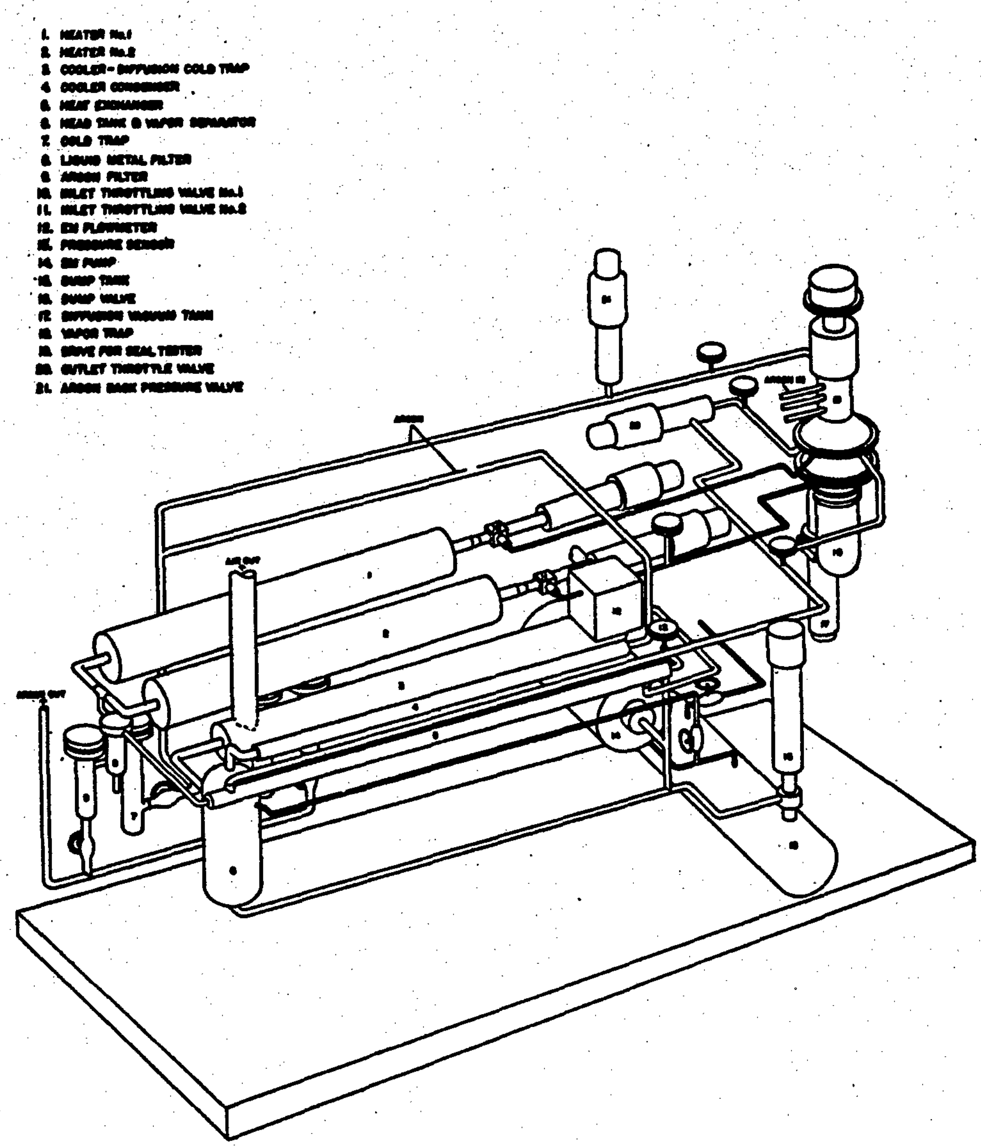

Figure 16. Seal Test Facility

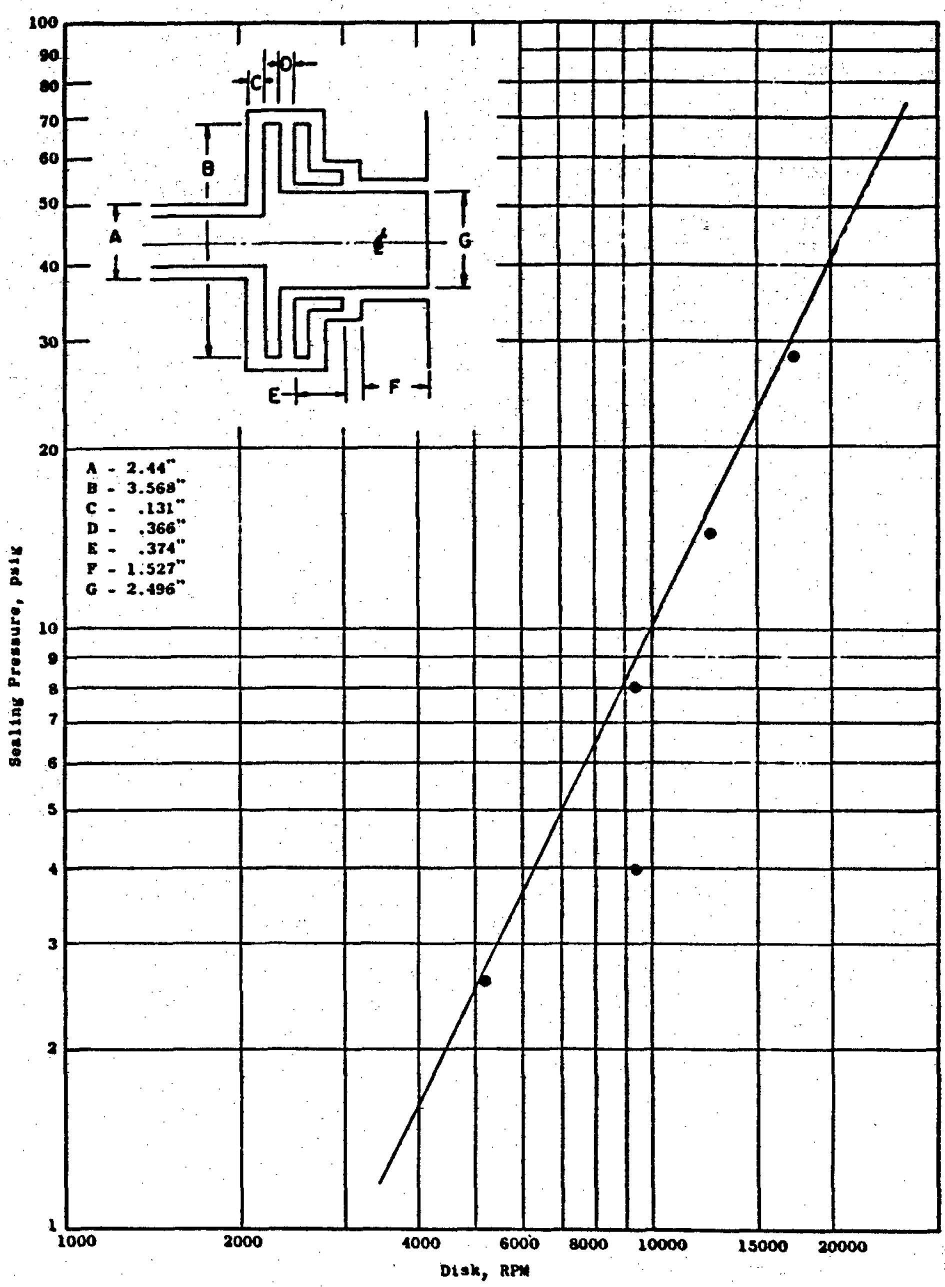

Figure 17a. Squeeze Seal Pressure Capacity, Configuration No. 3-20-1313.

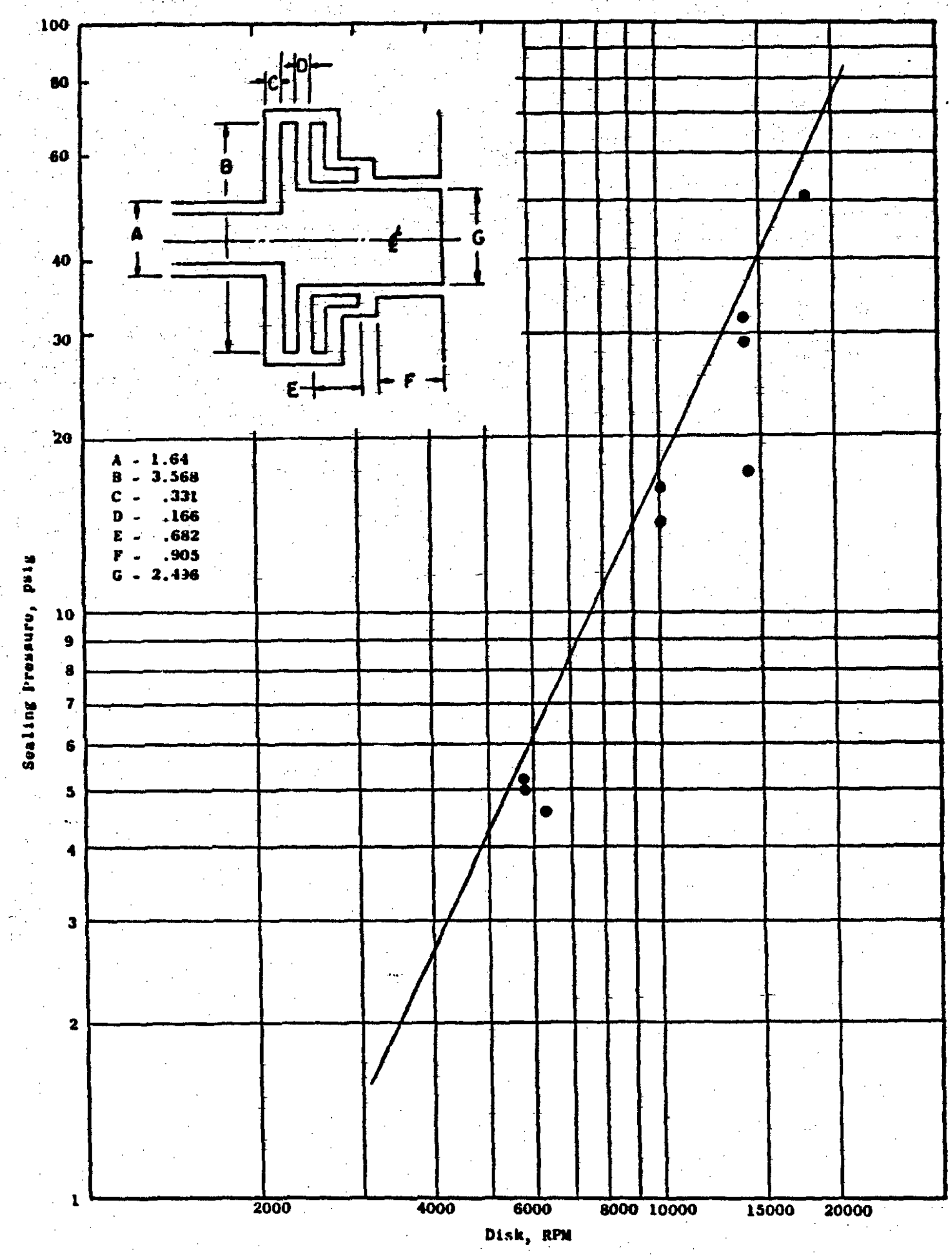

Figure 17b. Squeeze Seal Pressure Capacity, Configuration No. 3-32-2311.

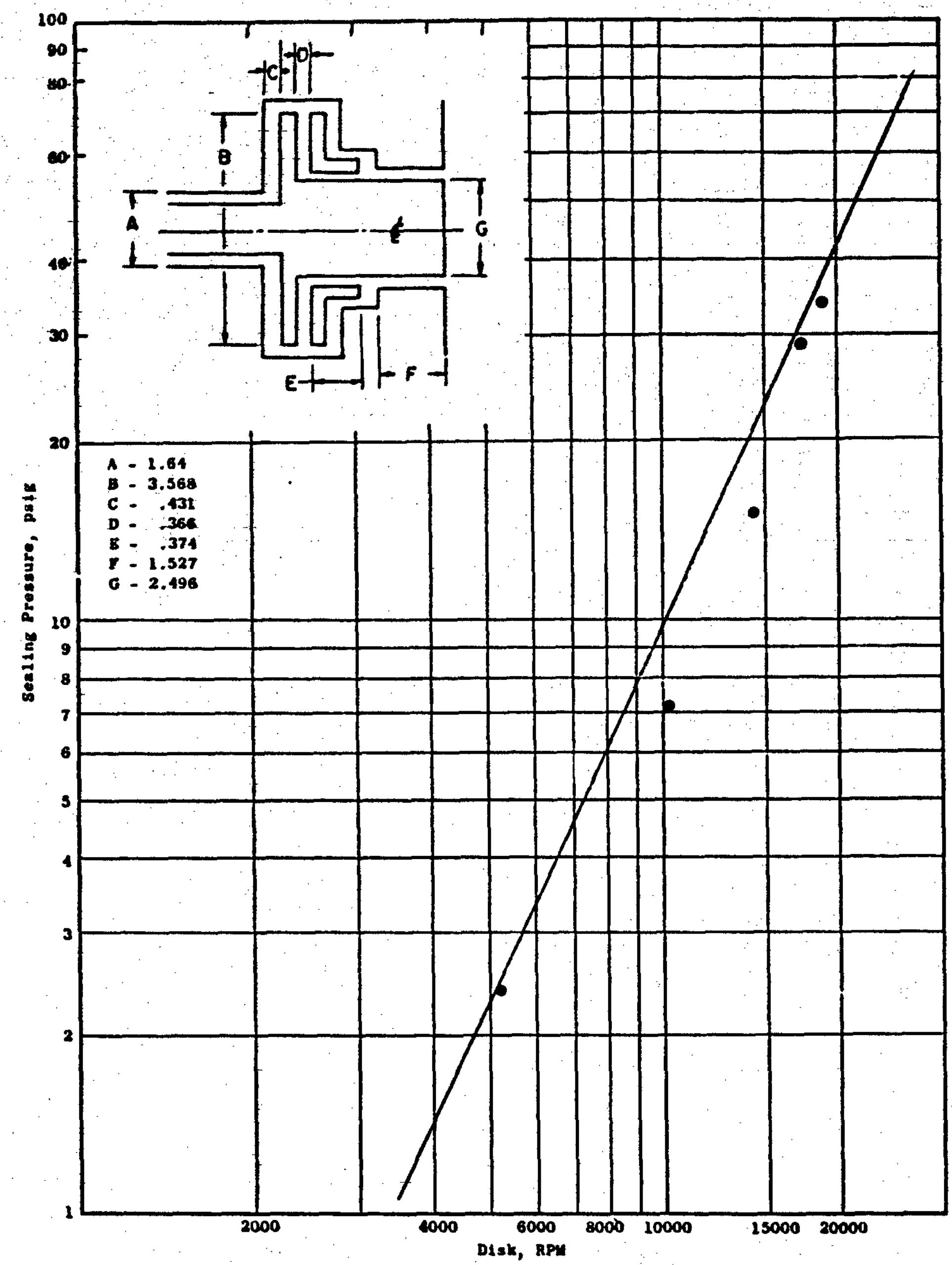

Figure 17c. Squeeze Seal Pressure Capacity, Configuration No 3-40-1313.

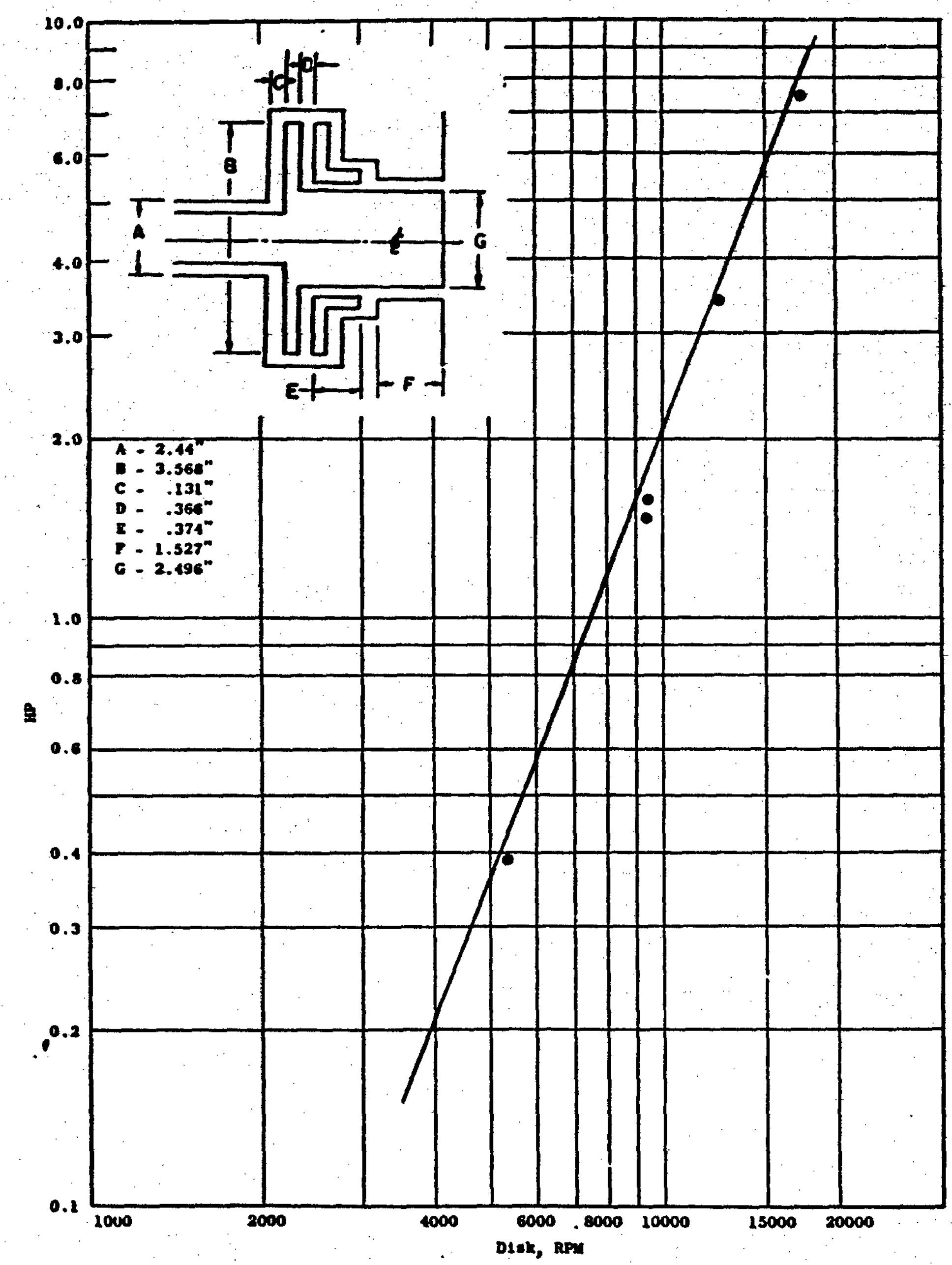

Figure 18a. Squeeze Seal Power Requirement, Configuration No. 3-20-1313.

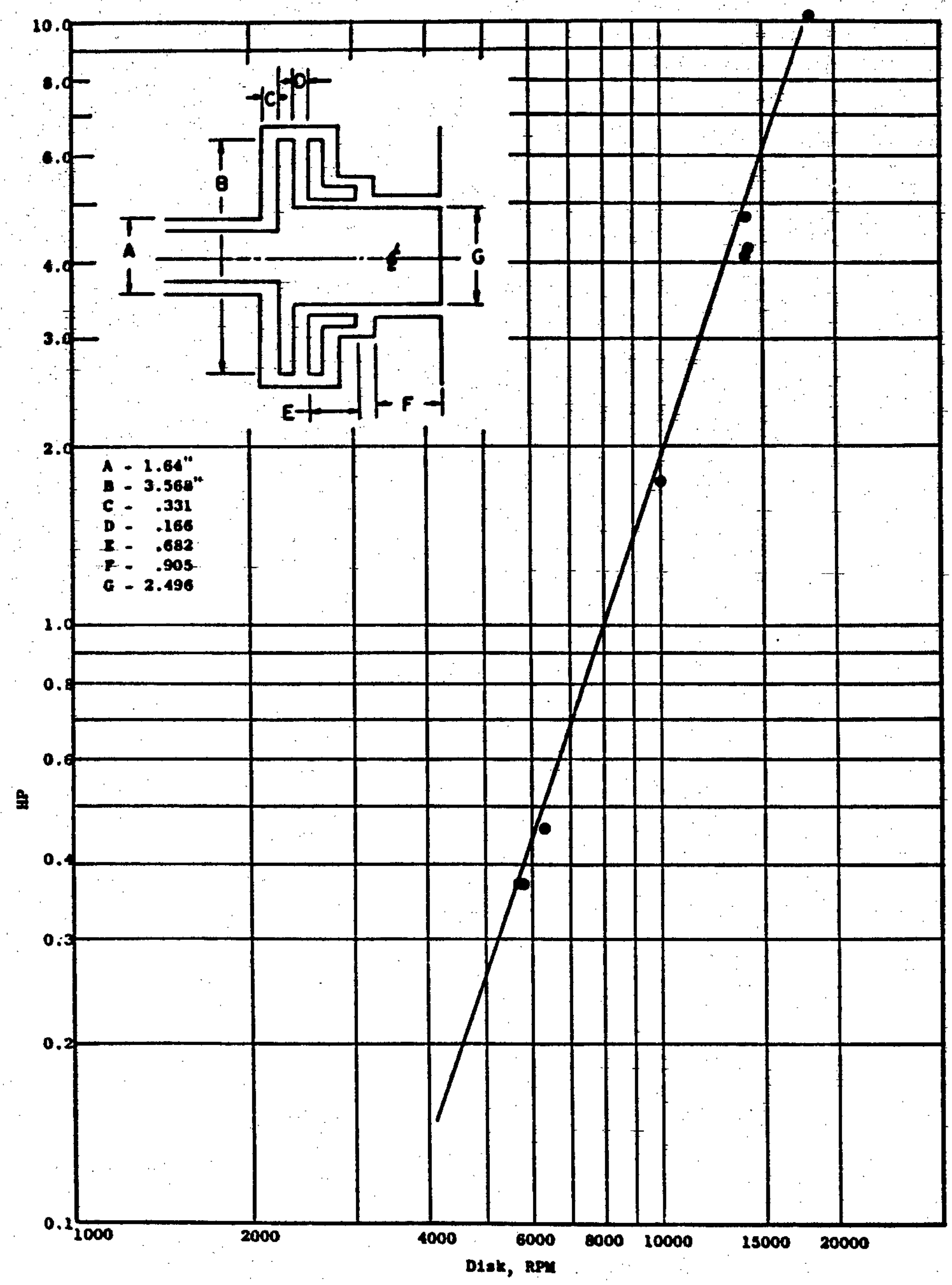

Figure 18b. Squeeze Seal Power Requirement, Configuration No. 3-32-2311.

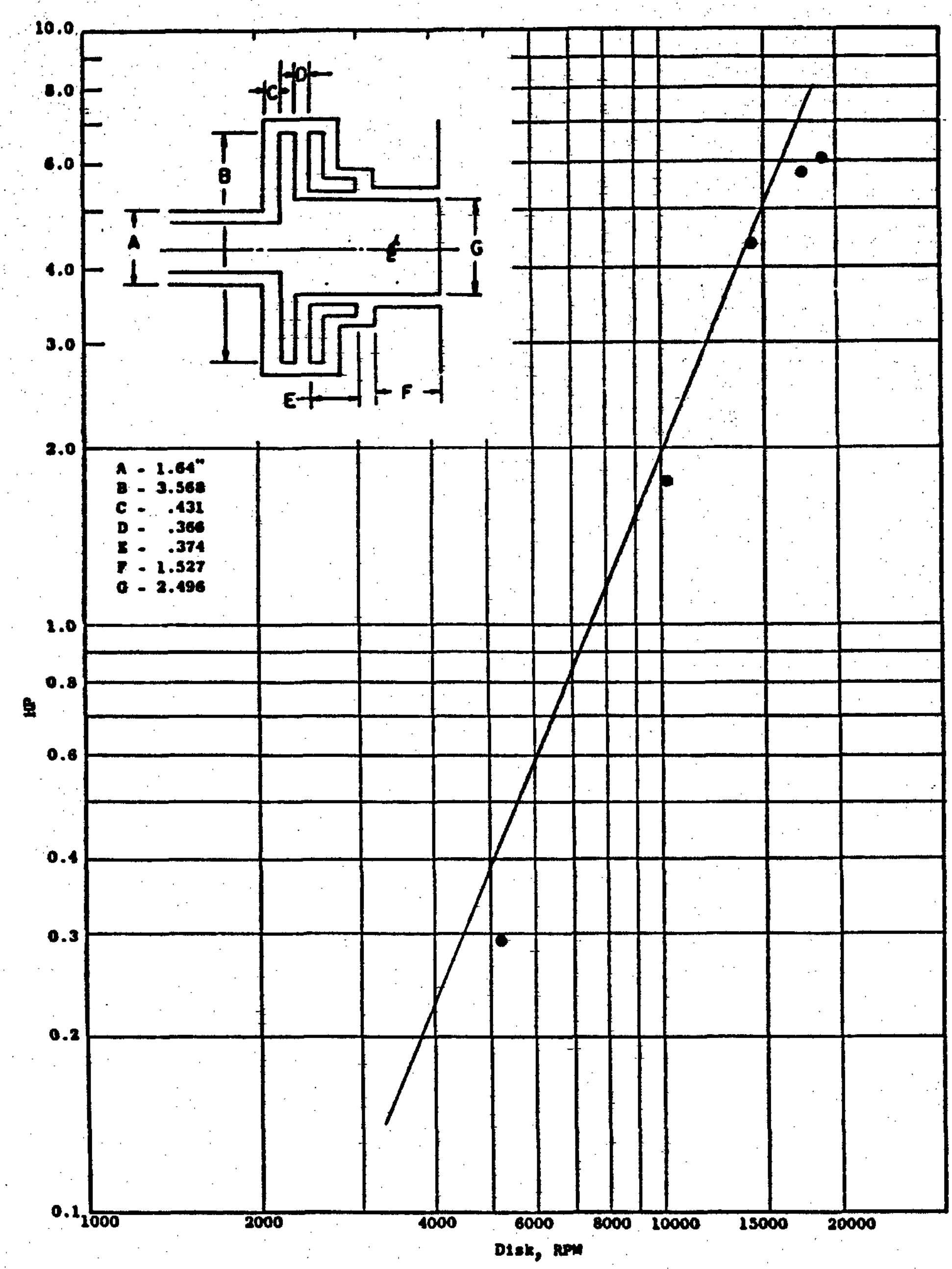

Figure 18c. Squeeze Seal Power Requirement, Configuration No. 3-40-1313.

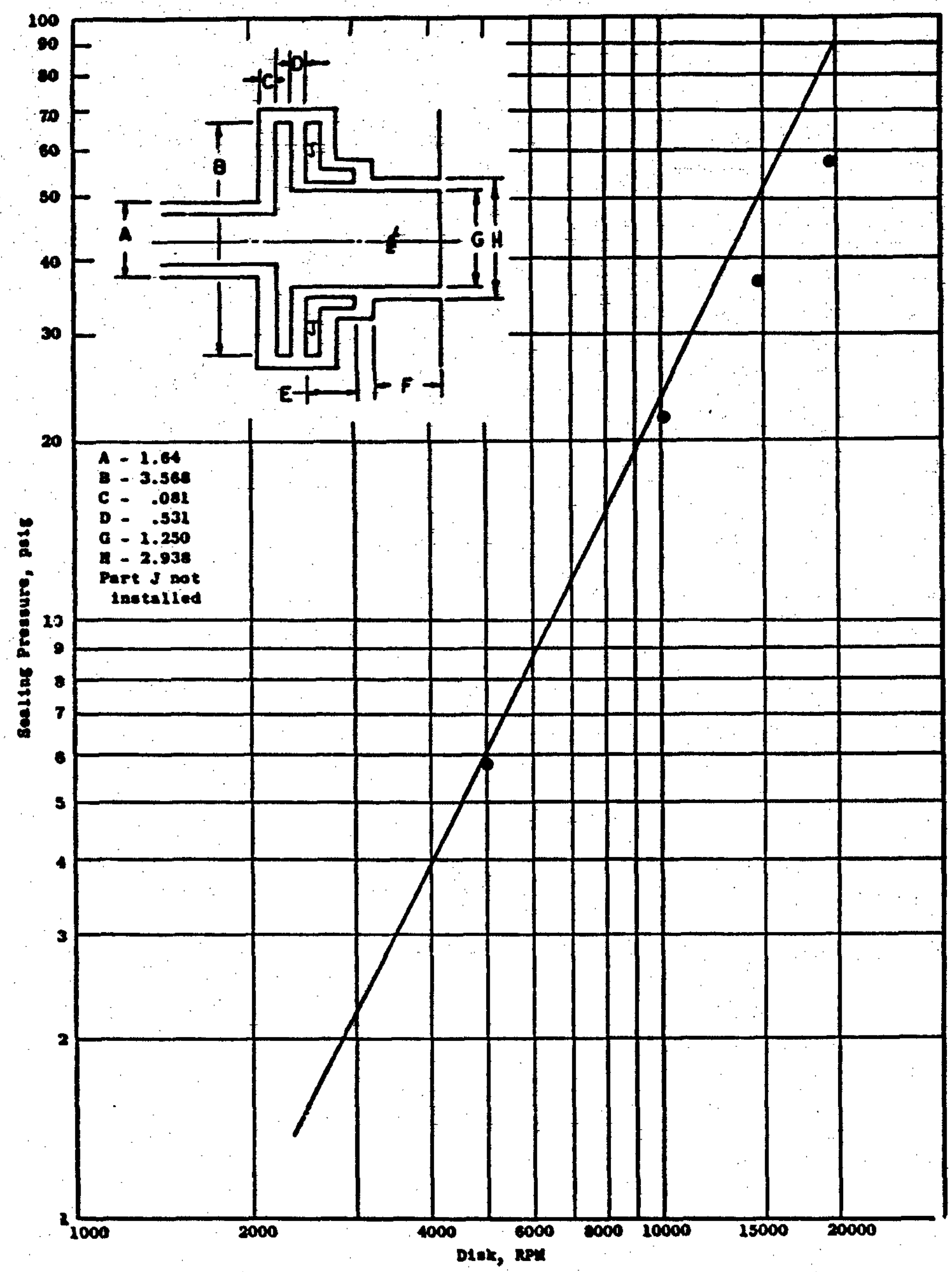

Figure 19a. Rotating Disk Seal Pressure Capacity, Configuration No. 13.

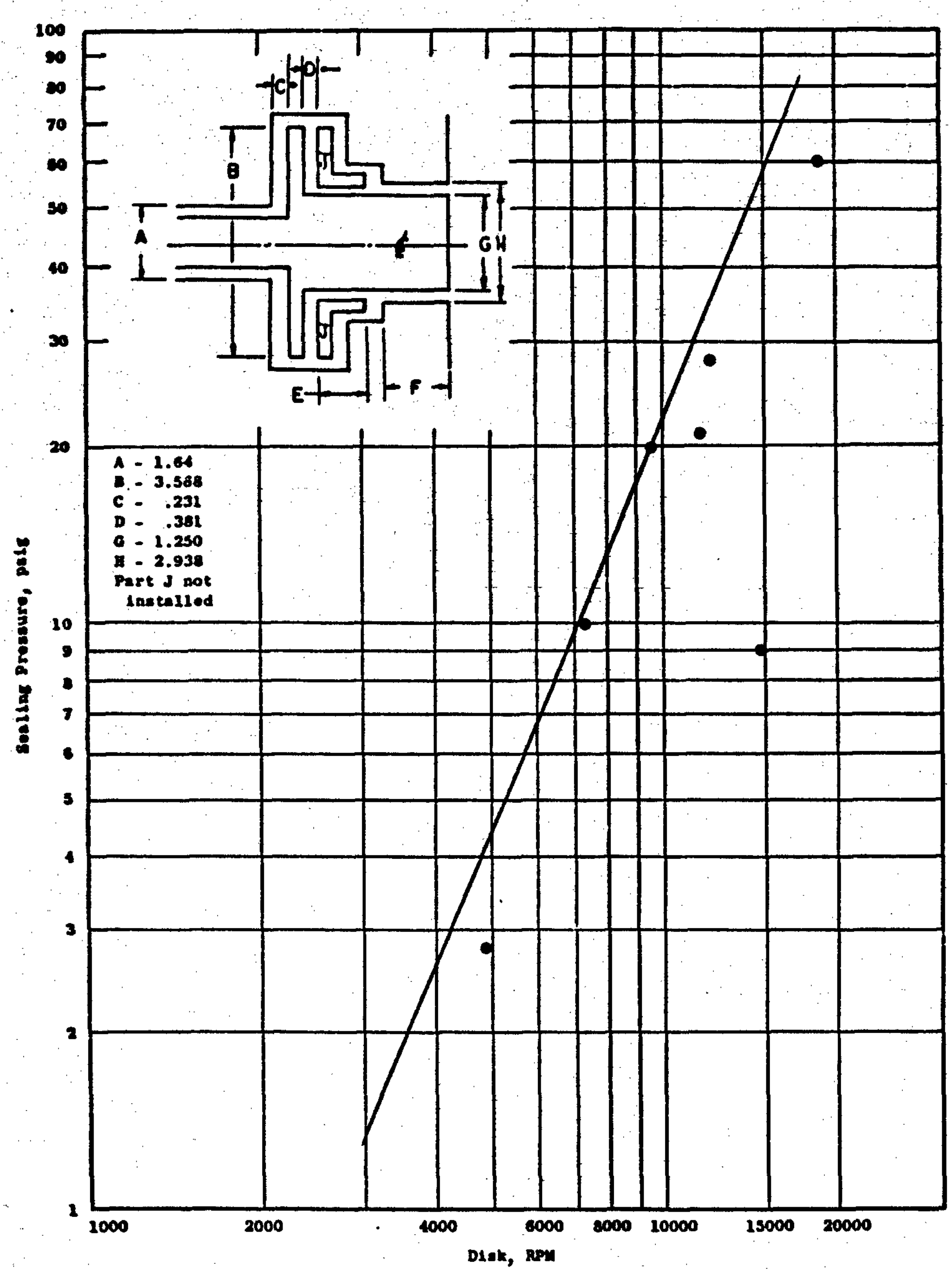

Figure 19b. **Rotating Disk Seal Pressure Capacity, Configuration No. 26.**

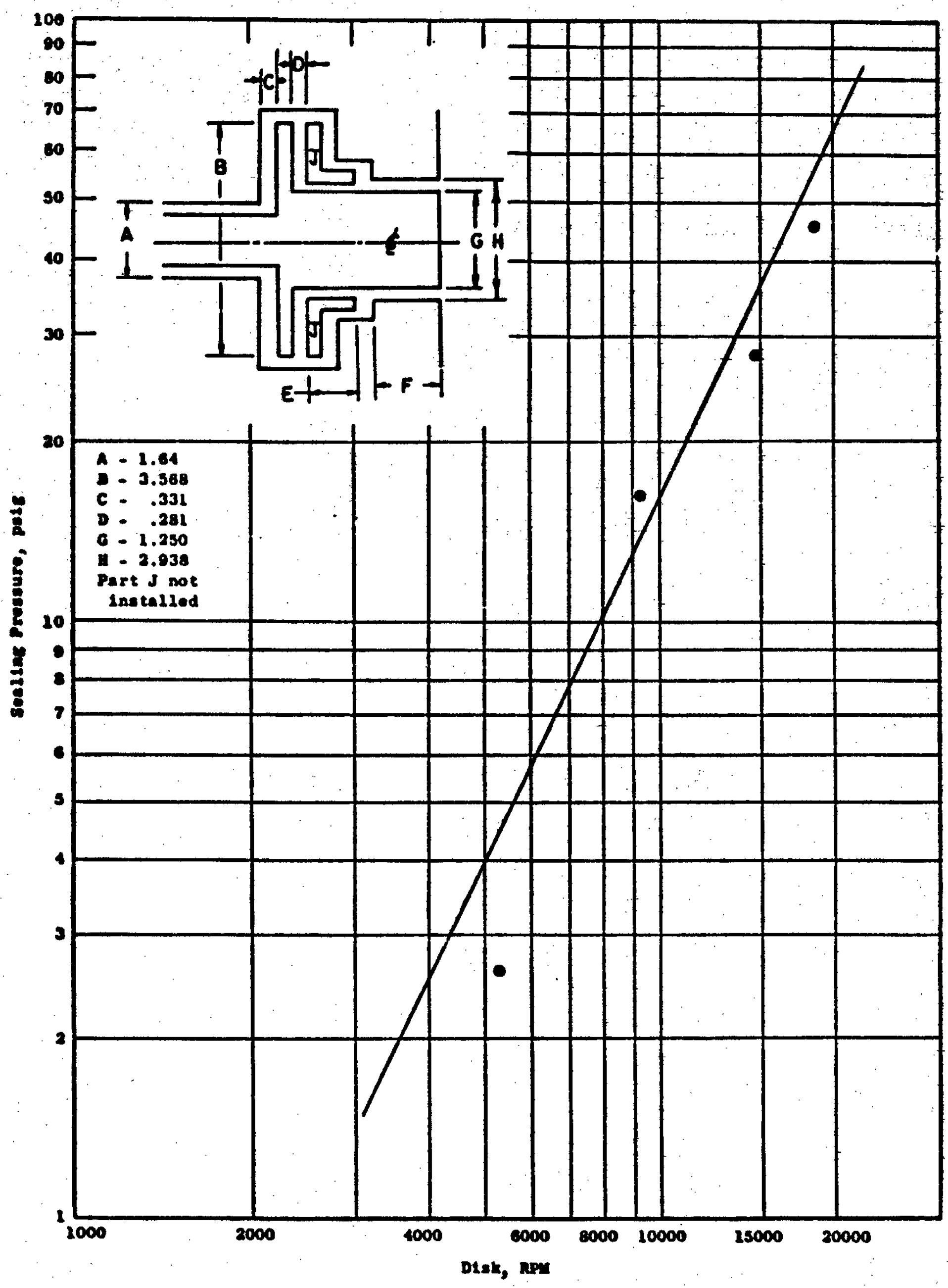

Figure 19c. Rotating Disk Seal Pressure Capacity, Configuration No. 32.

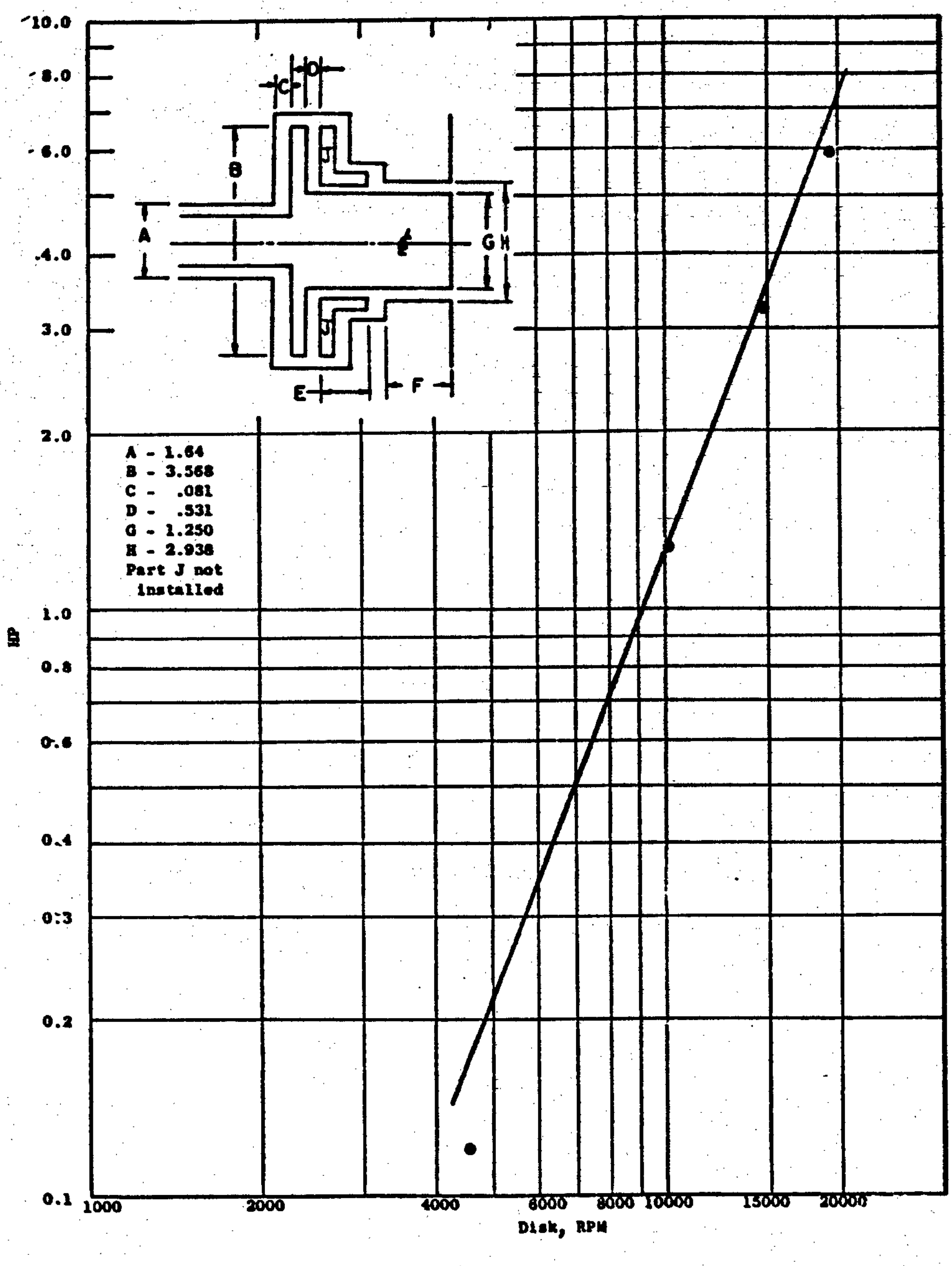

Figure 20a. Rotating Disk Seal Power Requirement, Configuration 13.

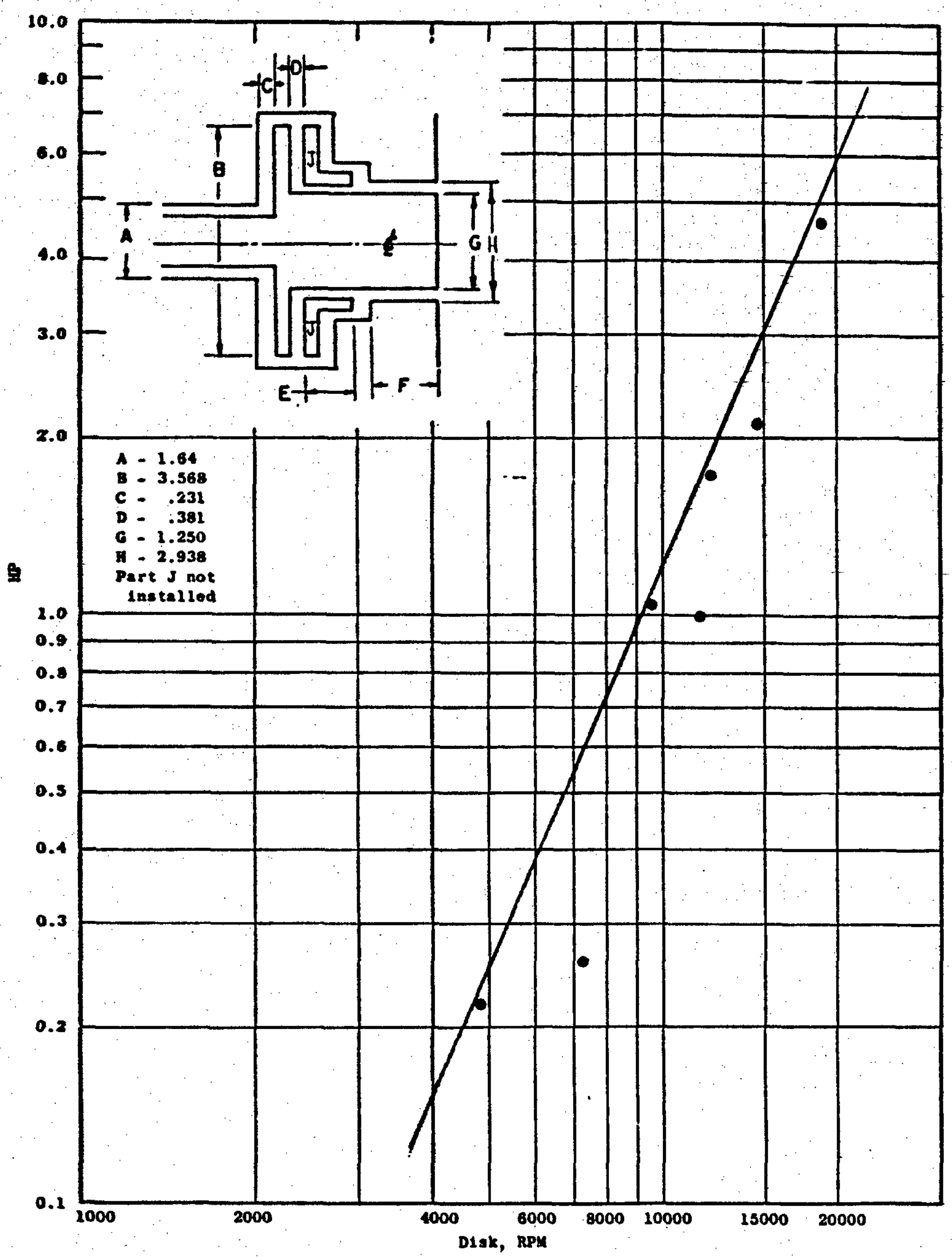

Figure 20b. Rotating Disk Seal Power Requirement, Configuration 26.

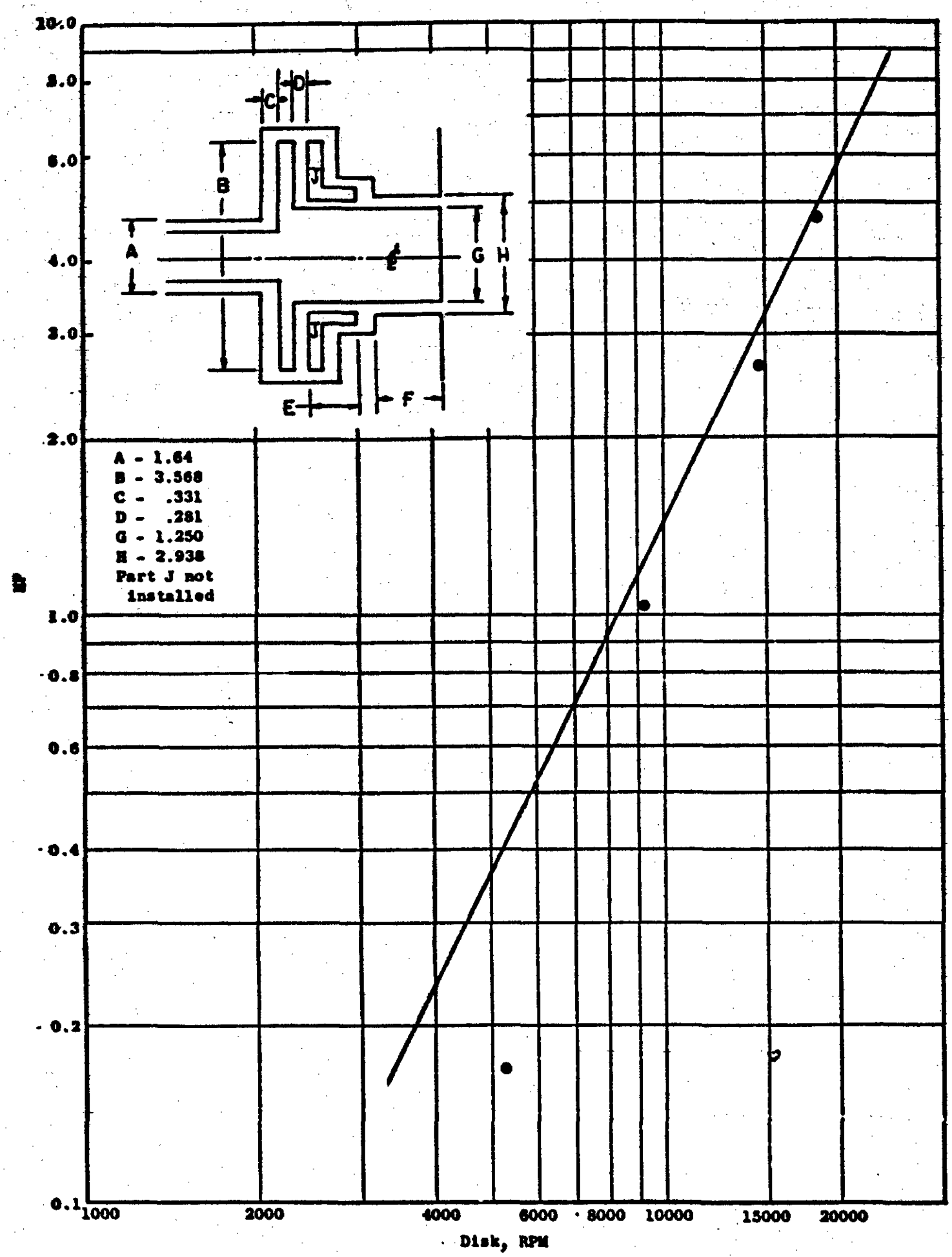

Figure 20c. Rotating Disk Seal Power Requirement, Configuration 32.

SUMMARY OF TEST RESULTS

Quill No. 1

Free Floating Sleeve

Test No.	Test Fluid	δ mils	N rpm	ΔP psi	Measured Power H.P.
1-1	SF96-5	3.50	3980	15.05	0.106
1-2	SF96-5	3.50	4760	20.34	0.113
1-3	SF96-5	3.50	6240	25.74	0.1043
1-4	SF96-5	3.50	4080	16.04	0.068
1-5	SF96-5	3.50	5660	20.54	0.1109
1-6	SF96-5	3.50	5700	25.54	0.1279
1-7	SF96-5	3.50	6940	30.54	0.1360
1-8	SF96-0.65	3.50	4560	4.5	0.064
1-9	SF96-0.65	3.6	6680	6.54	0.047
1-10	SF96-0.65	3.7	10360	10.24	0.1734
1-11	SF96-0.65	3.8	13460	13.14	0.2637
1-12	SF96-0.65	3.92	14940	15.44	0.2103
1-13	SF96-0.65	4.04	16560	18.24	0.1952
1-14	SF96-0.65	4.15	7080	6.14	0.0592
1-15	SF96-0.65	4.25	12160	10.44	0.1041
1-16	SF96-0.65	4.38	15400	15.64	0.2812
1-17	SF96-0.65	4.5	16840	20.44	0.2113
1-18	SF96-0.65	4.6	18760	22.94	0.3139
1-19	SF96-0.65	4.7	20420	25.64	0.4544

Figure 21

SUMMARY OF TEST RESULTS

Quill No. 1

Fixed Sleeve

Test No.	Test Fluid	N rpm	ΔP psi	L in	Measured Power H.P.
1-20	Distilled Water	5220	5.45	3.75	.0292
1-21	"	8560	10.05	2.80	.0479
1-22	"	11520	15.90	2.50	.0646
1-23	"	13600	20.55	2.42	.0761
1-24	"	15060	25.40	2.34	.0842
1-25	"	16340	30.40	2.31	.0915
1-26	"	6240	5.70	3.48	.0568
1-27	"	11780	15.65	2.51	.1169
1-28	"	17040	31.00	2.48	.1429
1-29	"	17860	35.65	2.29	.1498
1-30	"	19520	40.70	2.27	.1912
1-31	"	20660	45.5	2.20	.2023
1-32	"	21460	50.5	2.25	.2102
1-33	"	23080	55.5	2.25	.2418
1-34	"	24160	60.5	2.25	.2532
1-35	"	27760	57.0	1.57	.2909
1-36	"	29880	55.0	1.44	.3131
1-37	"	31520	55.0	1.23	.3303
1-38	"	18160	31.0	2.26	.0763
1-39	"	5600	7.0	4.00	.0392

Figure 22

SUMMARY OF TEST RESULTS (cont'd)

Quill No. 1

Fixed Sleeve

Test No.	Test Fluid	N rpm	ΔP psi	L in	Measured Power H.P.
1-40	Distilled Water	16520	30.6	2.36	.2080
1-41	"	35000	56.8	1.21	.3383
1-42	SF95(5)	1360	5.3	4.00	.0764
1-43	"	3020	10.7	2.79	.1902
1-44	"	4520	15.5	2.53	.3793
1-45	"	6120	21.0	2.44	.5562
1-46	"	7780	26.0	2.37	.7622
1-47	"	9320	31.0	2.31	.1108
1-48	"	10700	36.1	2.28	.1272
1-49	"	11640	41.2	2.28	.1384
1-50	"	12700	46.2	2.27	.1598
1-51	"	13580	51.2	2.26	.1709
1-52	"	14320	55.8	2.24	.1803
1-53	"	15420	61.0	2.24	.1942

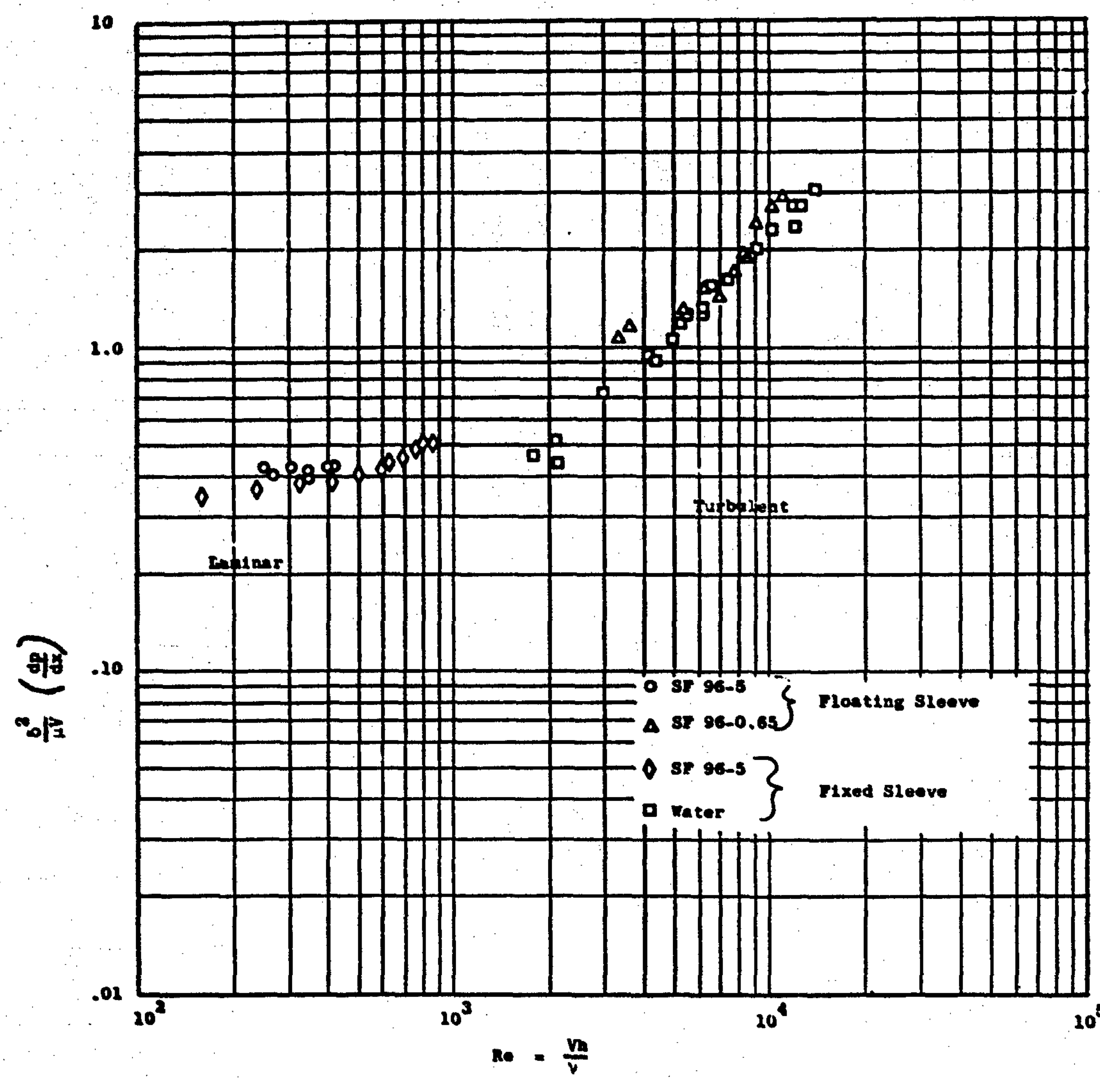

Figure 23. Sealing Coefficient vs. Reynolds Number.

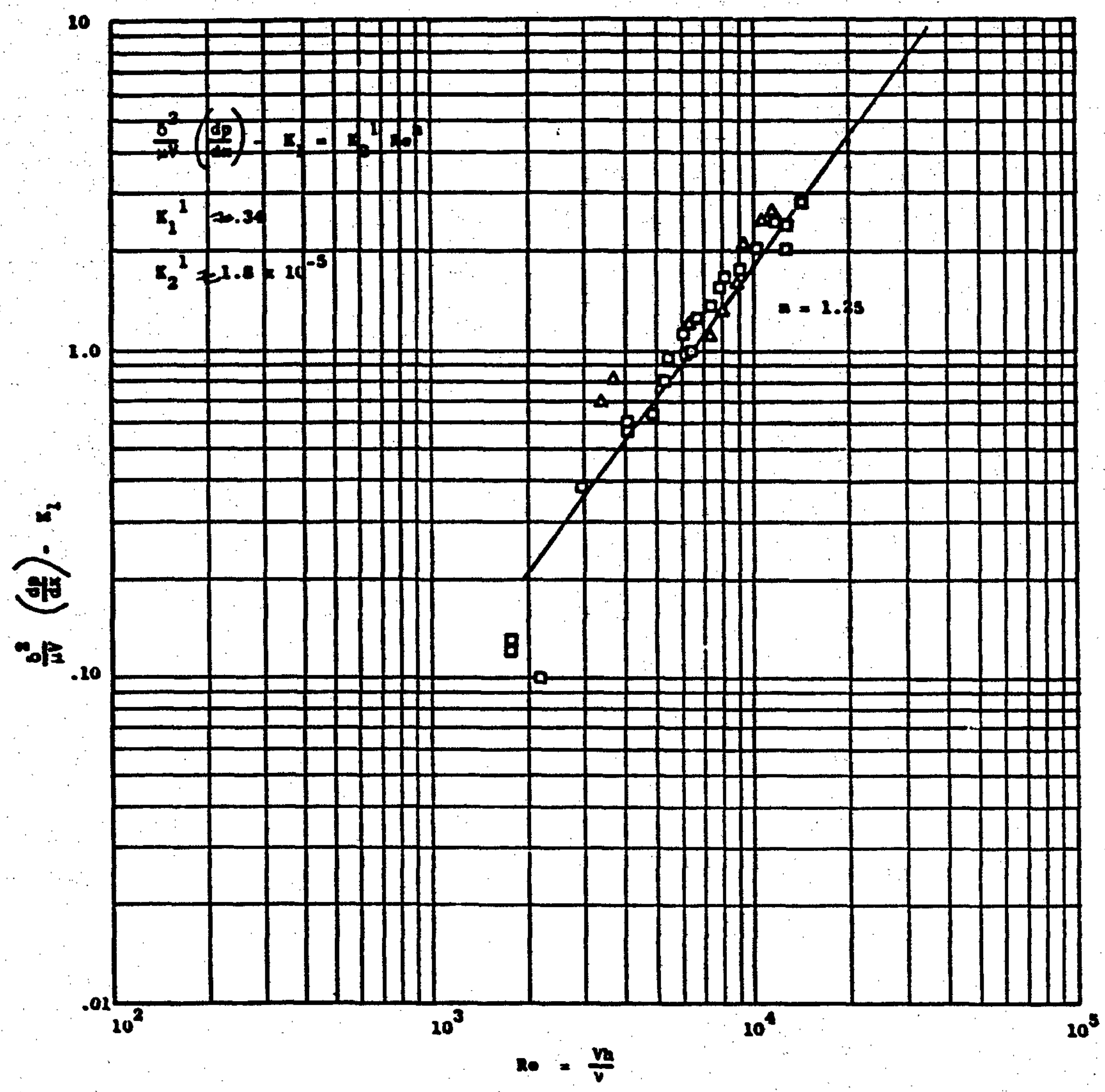

Figure 24. Sealing Coefficient vs. Reynolds Number.

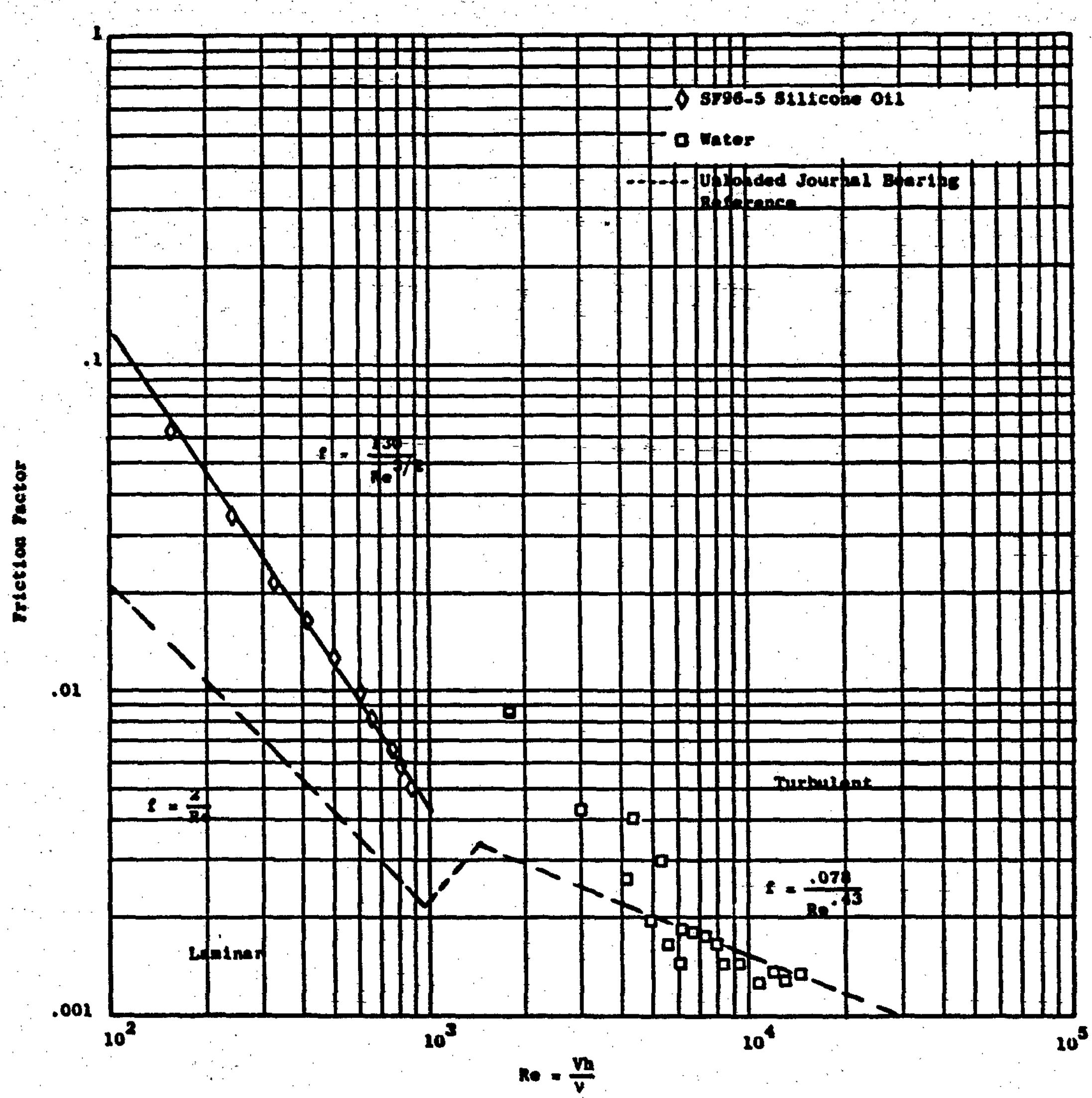

Figure 25. Screw Seal Friction Factor vs. Reynolds Number.

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8. Quarterly Report for Period Ending July 15, 1962, "Dynamic Shaft Seals in Space," Contract AF 33(657)-8459.

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